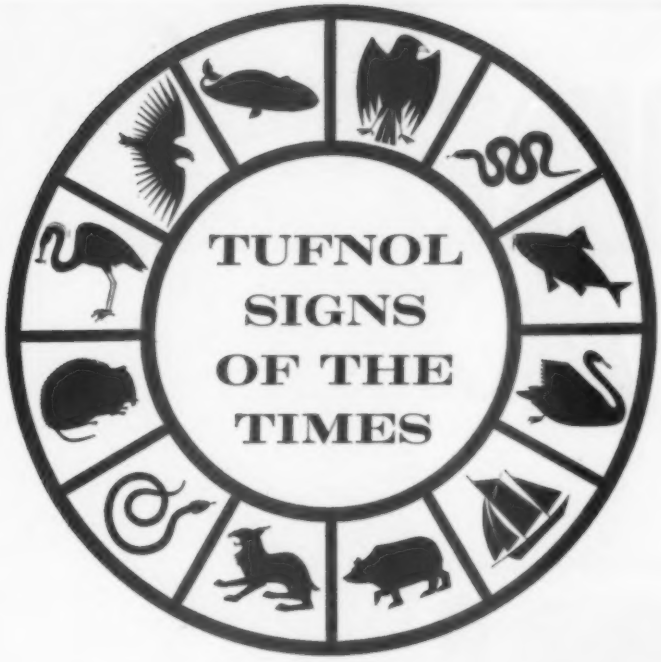
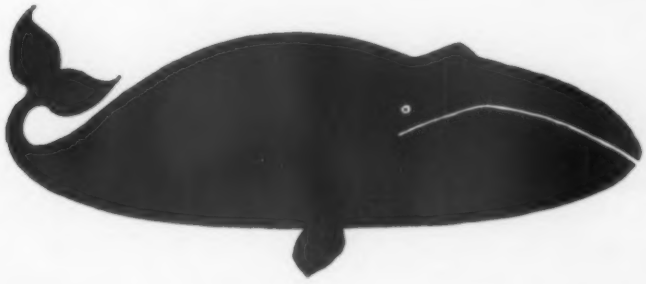
AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 51 No. 8 AUGUST 1961 PRICE: 3s. 6d.



Whale-sign of Tufnol



Those people born under the sign of the Whale have something to spout about. They become chief purchasing officers, chief engineers, chief draughtsmen and chief designers. Whale men leave it to Jonah's disciples to wail about the inadequacies of the materials they have at hand. Whale men are perfectionists who travel far when they find one material that is light yet strong and hardwearing, resistant to corrosion and a good electrical insulator. They also find that this ideal material resists deterioration in storage, machines easily with ordinary tools, and is available in sheets, tubes, rods, angles, and channels. Many Whale men have already discovered this remarkable material—"WHALE" brand Tufnol. They have also discovered that Tufnol is available in eleven other brands, each with pre-determined properties.

All brands of Tufnol are laminated plastics; each formulated to satisfy specific requirements. If your birthday falls between January 1st and December 31st this could be your lucky week—the week you discovered Tufnol.

'Phone or write to your local Tufnol Branch Office and we predict that you will thank your lucky stars.

TUFNOL
REGD. TRADE MARK

Tufnol Limited,
Perry Barr, Birmingham 22B

P3

One tool does the work of two!

One operator with this Desoutter tool can do *two* jobs; he can tap holes and run studs, and he can do both jobs quicker, easier and cheaper. The Desoutter tapper-studrunner is just what you want on one-off assembly work. It works on the push-pull principle; when you pull it out it reverses automatically at twice the forward speed. And it has a quick-release chuck that lets you change over from tapper to studrunner in seconds. There's a type, size and weight for the job *you* have in mind. Ask us for further details.

TYPE	R.16/75	R.16/40	R.16/25
TAPPING SPEED	900	460	315
REVERSED SPEED	1,800	920	630
CAPACITY IN STEEL	$\frac{1}{4}$ in.	$\frac{5}{16}$ in.	$\frac{5}{16}$ in.
	6 mm.	8 mm.	8 mm.
WEIGHT	$4\frac{1}{2}$ lb.	$4\frac{1}{2}$ lb.	$4\frac{1}{2}$ lb.
	2.0 kg.	2.0 kg.	2.0 kg.

DOUBLE YOUR MONEY WITH *Desoutter*

DESOUTTER BROTHERS LIMITED
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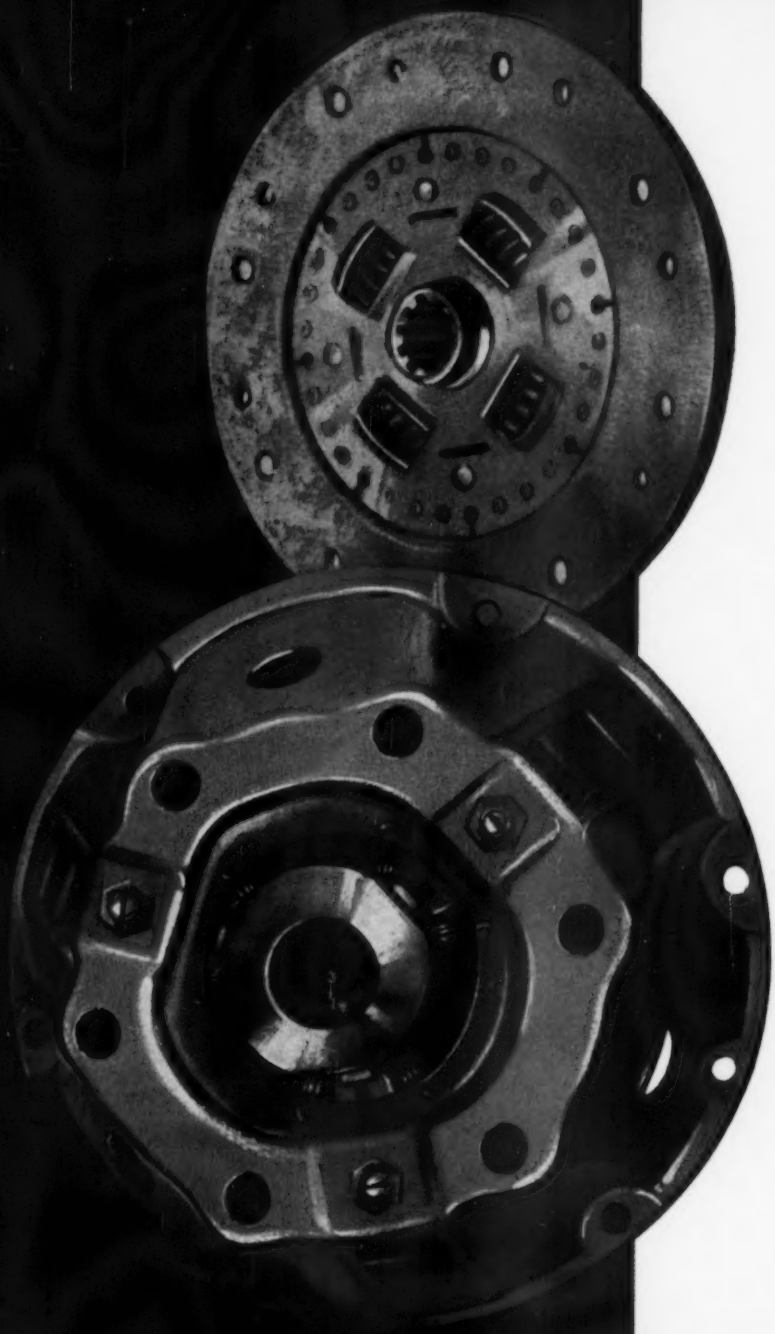
GBC348

Clutches by BORG



Borg & Beck clutches have ac
by a combination of brilliar
excellent workmanship.

Basic features include uniforr
pressure-plate of great rigidit
release-lever system having th
centre cushion drive plates wit
a great deal to smooth the trar

The illustration at left shows
these are made in the follo
10", 11".

The strap drive, shown in the
clutch, permits the pressure-p
to-metal sliding contact. It wa
clutches and its success has l
'extended range is now as foll
and larger sizes on applicatior

In addition, the 18" R.4 hea
single or twin-plate type.

For tractors and other equip
Borg & Beck clutches, enab
for both propulsion and drivi

In addition there is a wide ra
clutches and power take-offs.

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ONE OF THE AUTOMOTIVE PRO
contributions to the Br

& BECK

REGD. TRADE MARK

eved their sweeping success
design, fine material and

spring pressure on a heavy
and heat capacity, with a
minimum friction. Spring
built-in dampers have done
mission of the modern car.

ne of the 'A'-type clutches;
ing sizes: 6¼", 7¼", 8", 9",

ght-hand picture of the 12"
te to travel without metal-
irst introduced in the larger
en so outstanding that the
ws: 8/8½", 12", 13", 14", 15",

-duty clutch is available in

ent there are two two-way
ng the one engine to serve
equipment or implements.

ge of Rockford over-centre

OMPANY LTD.
VICKSHIRE, ENGLAND

DUCTS GROUP'S
tish Motor Industry . . .

EASIER DRAWING AND PRESSING...

Photograph by courtesy of Lec Refrigeration Ltd

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ELECTRO-ZINC COATED SHEET STEEL

DRAGONITE is sheet steel which has been given a coating of pure zinc on both faces. It has a number of advantages over ordinary, uncoated sheet steel. During drawing and pressing operations, for instance, the surface properties of Dragonite are maintained. In fact, the fine-grained structure and natural ductility of pure zinc actually *assist* fabrication by acting as a dry, solid lubricant.

This ductility of Dragonite cuts manufacturing costs because it makes longer press runs possible and lengthens tool life.

There are many more good reasons why you should be using Dragonite. For fuller details, please write for a copy of the *Dragonite Technical Handbook* to:

These are some of the industries in which Dragonite is being used extensively: Office Equipment; Elevators; Agricultural Equipment; Automobile Industry; Domestic Appliances, Radio Equipment, etc.

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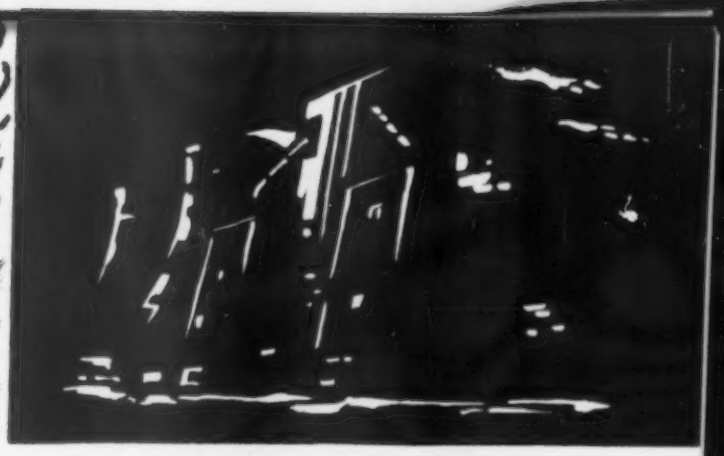

New projects of science and industry,
improved methods of production and
new materials to be worked,
have brought a challenge—
a challenge to provide cutting tools
equal to these new demands.
A challenge met by DORMER.
Our experience, technical knowledge
and constant research enable us
to keep up with modern developments,
and our methods of manufacture
are co-ordinated to maintain
our position in the forefront.
DORMER TOOLS consistently answer present-day needs, and anticipate those of the future.

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SEPTEMBER 3—12

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The impressive new Ford will earn enthusiastic compliments from trade and public alike. No manufacturer is more exacting than

FORD

in the choice of components. The Silentbloc Group take it as a very great compliment that so many of their products are incorporated in this brilliant new model.

SILENTBLOC AND **ANDRE RUBBER**

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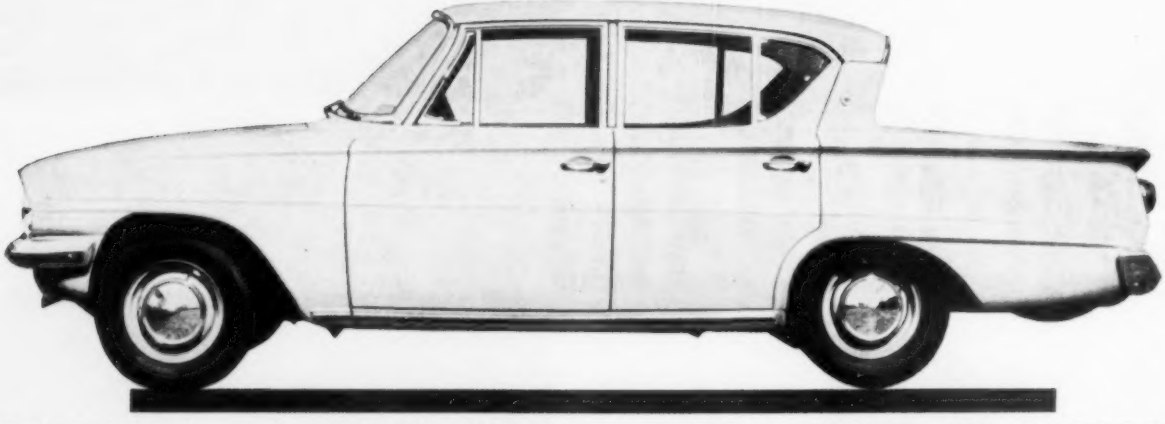

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(roadway/513)

NYLON AT ITS BEST IS CALLED...

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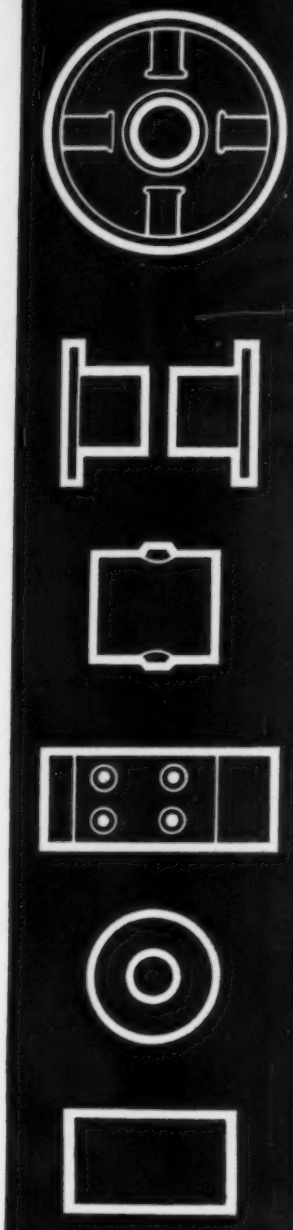

The Nylastic bearing is now firmly established in the Automotive world as the ideal bush for many swivel arm and lever applications. Nylastic is also used for track-rod ends, diesel fuel pump coupling discs, chain reaction pads, cable support clips, thrust washers etc., and is used with combinations of rubber and metal for steering column bearings, anti-vibration clips, earthing clips etc.

The experience gained over the past years enables us to give a unique service in advice and design for this increasingly popular material. Our Technical Staff is available to give full information about Nylastic, and to produce drawings and suggested schemes for your particular applications.

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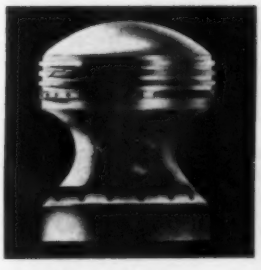
BHB

what does it mean?

So often this question is asked. When, nearly 20 years ago Bentley Motors started producing Motor Cars, the hour glass type of piston proved to be so noisy that a new design was developed and patented. The Patent was granted in the names of three eminent engineers, and the first letters of their names became the symbol of this *famous piston.*



Manufactured by

Automotive Engineering Limited

(One of the Sheepbridge Engineering Group)

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— assembled to customers drilling requirements

Herbert All-electric Drilling Machines are made in ten types for drilling holes from the very smallest, at 18,000 r.p.m., up to 1¼" diameter.

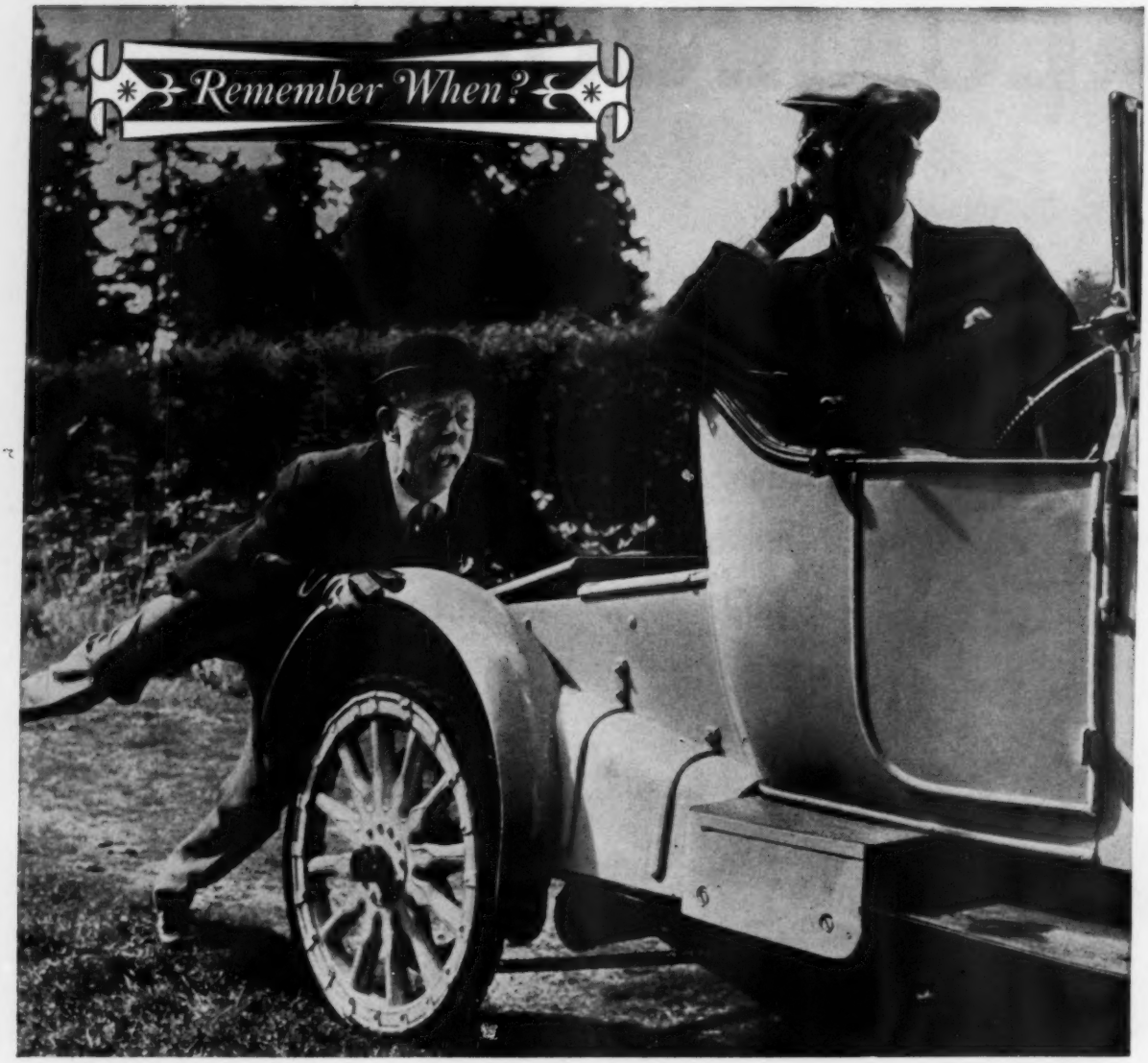
This equipment meets all general-purpose single or multi-spindle requirements. Unit construction and interchangeability enable special requirements to be easily met.

We will quote for machines and equipment to meet special requirements for drilling, tapping and associated operations.

Three Type M Multi-spindle top columns on a common base for drilling seven ·328" dia. holes and drilling and tapping five ⅜" dia. 26 t.p.i. holes in a gearbox inner cover. Cover located in traversing fixture. Floor-to-floor time is 3 minutes.

ALFRED HERBERT LTD., COVENTRY

AD. 507



Those difficult moments. Aching back. Sore hands. Strong words! Doubts begin to occur. Internal Combustion begins to seem an infernal mistake.

Doubts? You can't avoid them even today. So it's good when something comes along with *no* doubt involved. Like the choice of BP Diesel.
BP Diesel is available at Agency sites throughout Britain. And with a Diesel Agency Card issued by Shell-Mex and B.P. Ltd your drivers can fill up with BP Diesel on credit, or cash at agency rates.

Philosophical Pastry at the wheel, and Powerful Pastry at the rear of the 1913 *Trojan*. One of the only two surviving prototypes of the 'Motor-car for the Millions', the *Trojan* is on display at the Montagu Motor Museum.

PETROL AGENCY DERV

< THIS IS THE SIGN TO LOOK FOR

BP DIESEL

Talking about Production Tools Are you needing a single tool for a special job . . . or several hundred for a big production programme? You will find *all* the answers to your problems in the CP range. Every compact and powerful CP tool has been fully proved in tough service conditions like the new 3019 Nut runner shown above. It's reliable and efficient, designed for its job, and saves you money every moment it is used. If you want to put new impetus into production, switch to CP tools and watch work output soar! *Literature is available on request.*

✳ *REVERSIBLE SCREWDRIVER AND NUTRUNNER (CP 3019-ORSC-2000)* Features the adjustable ratcheting clutch. Screw tightness or torque varied by adjustment of tension of a spring which holds ratcheting jaws together. Provides impact tightening action at desired degree of torque, clutch jaws then engage and disengage until tool is removed. *Capacity: ¼" Machine Screws, No. 6 wood screws.*

Consolidated Pneumatic

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CT-65

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-the first major development in belt drives in 60 years

- Greater power transmission ▪ Less power loss
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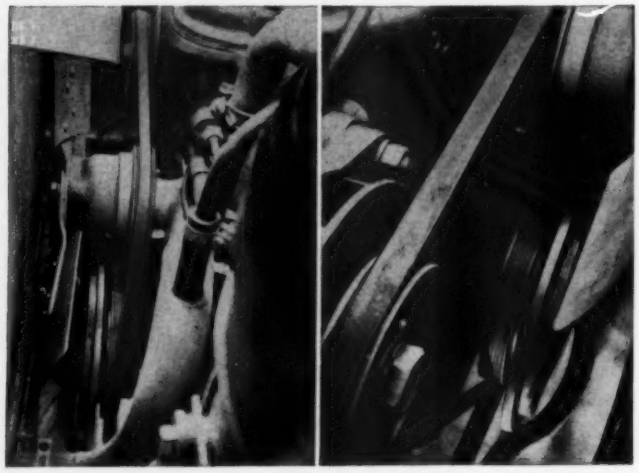


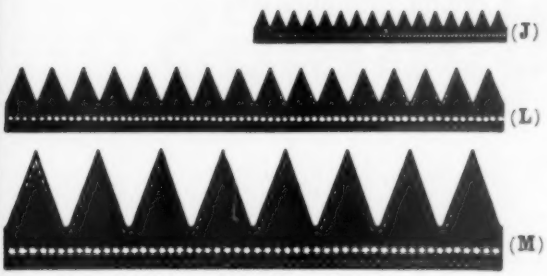
POLY-V belts are an entirely new concept in transmission drive . . .

POLY-V* belt's multiple ribs grip pulleys throughout the entire friction surface—a surface twice the area of conventional multi-drives, one third more than conventional single drives.

Power wastage, slip and abrasion are reduced to minimal proportions. Because of the higher ratios made possible, transmission systems can be simplified and reduced to minimum size, weight and cost. A far more compact drive unit can be achieved and the shallow grooves permit easier fitting and replacement.

POLY-V belts are made from synthetic rubber compounds (having the heat-and-oil-resistant properties of today's "Premium" V-Belts). Strength is given by high-tenacity synthetic cords. The base rubber is 'soft' compared with orthodox 'V' Belts—its action resembling a hydraulic fluid and forcing the rib into perfect mating with the pulley grooves, with even overall pressure.

POLY-V BELTS NOW available in Britain!



THERE ARE THREE BASIC RIB SIZES OF POLY-V BELT

Section	Rib Width	Rib Quantities	
		Minimum	Maximum
J	$\frac{3}{32}$ in.	2	16
L	$\frac{3}{16}$ in.	6	20
M	$\frac{3}{8}$ in.	4	20

V angle is 40° in each case.

SPECIAL APPLICATION OF "POLY-V" BELTS FOR THE AUTOMOTIVE INDUSTRY

The advantages of POLY-V belts have been amply proved in America—where they are now extensively used throughout industry.

In the design of engines for heavy public service and commercial vehicles, POLY-V drives can replace the multiple systems used for fan, water pump and generator drives with a *single belt and pulley system*—and transmit the same horsepower from narrower pulleys eliminating the necessity to fit matched belt seats.

POLY-V belts can give neater, more efficient drives for air-conditioning units, supercharger and tachometer drives and pump units on power steering equipment. Further usage may be for dynamo drive in motorcycles—perhaps even as a direct drive for mopeds and lightweight motorcycles.

POLY-V
TRADE MARK

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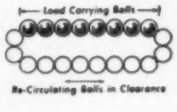
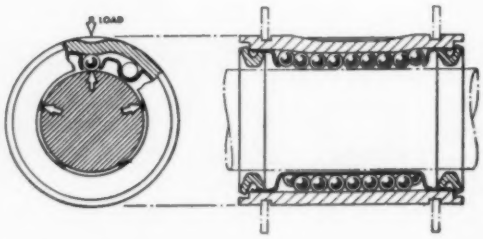
Ball Bushings are fully described in **R & M** publication 76. If you have linear motion, then you need a copy.





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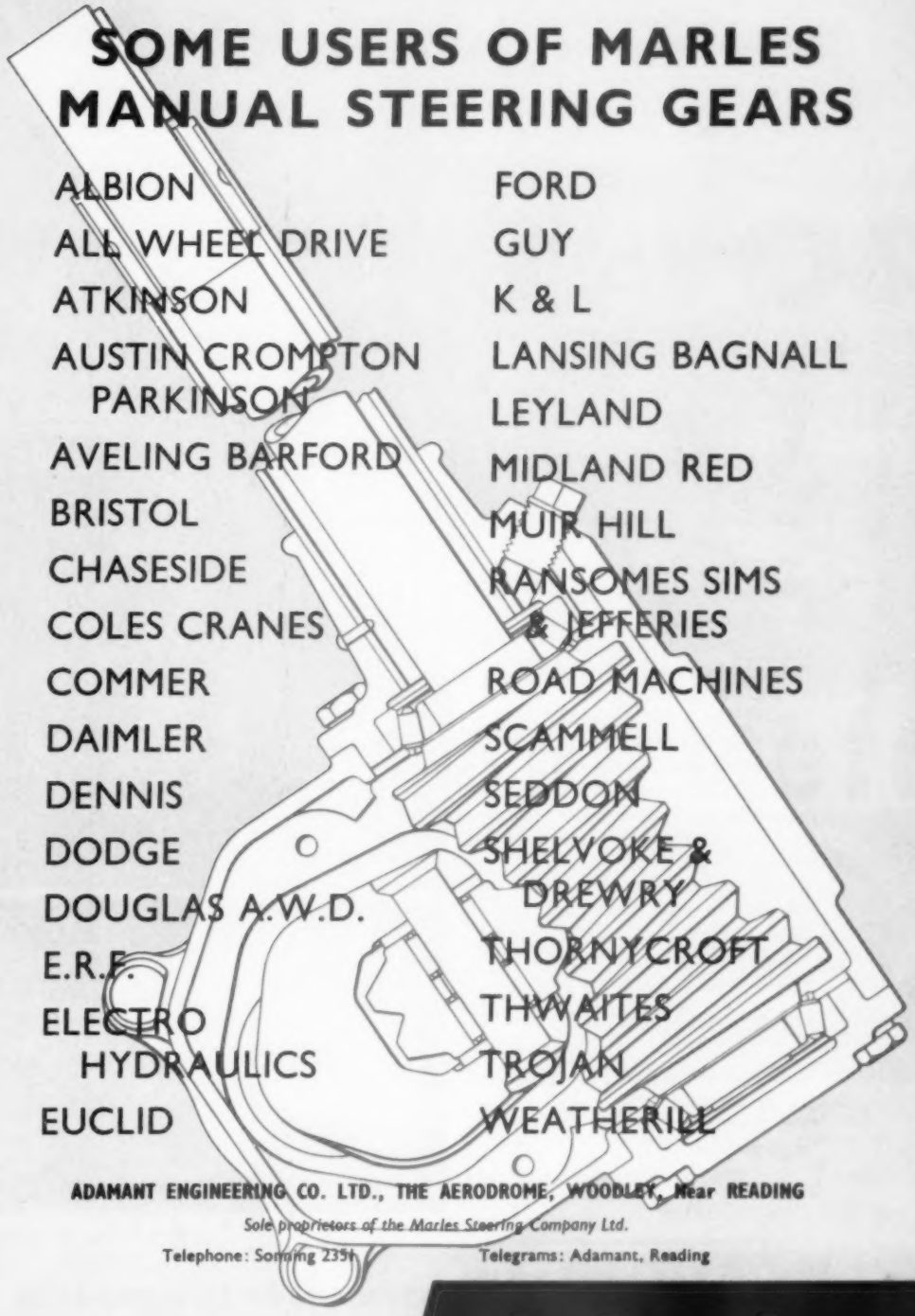

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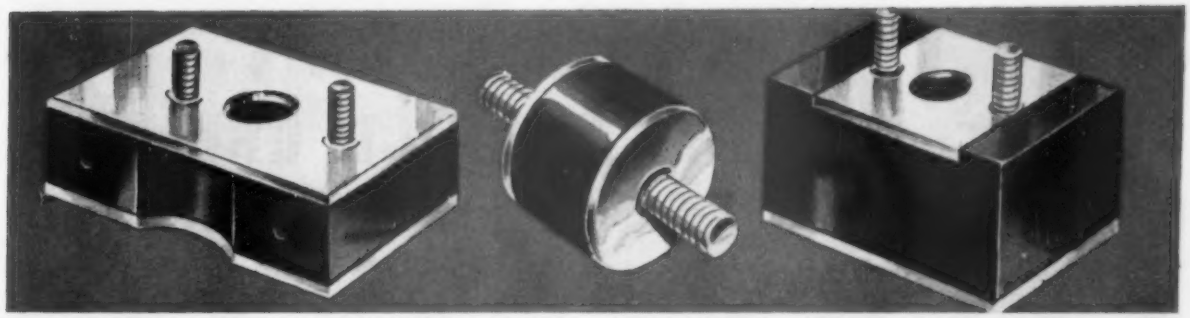
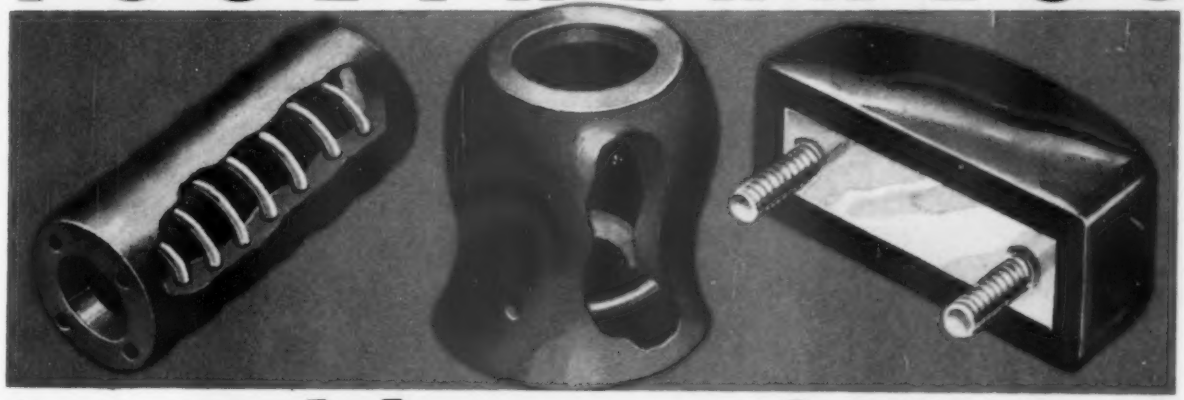
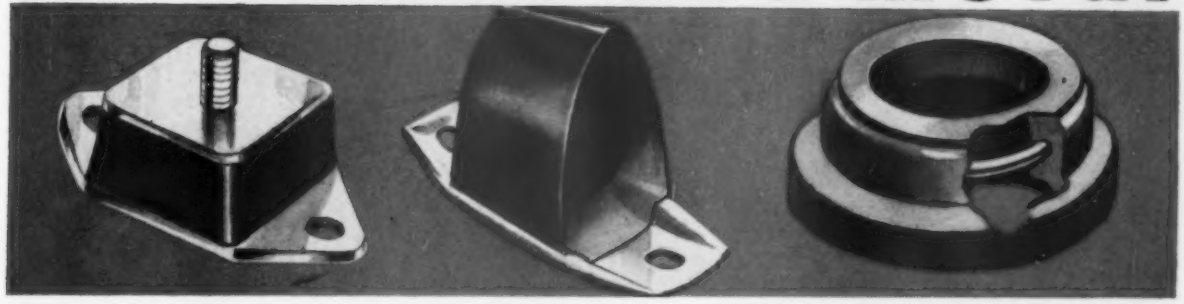


TOGETHERNESS



in rubber and metal



Specialists in the technical rubber field, we manufacture both the rubber and the metal parts and then effect the bonding — all in our own factory. The result ? Long lasting unions under all sorts of conditions.

Precision moulding, of natural and synthetic polymers and resins, can be carried out in our factory to tolerances of 0·001" or 0·002", according to requirements. Our technical, scientific and design staff also carry out experimental work on new designs and applications.

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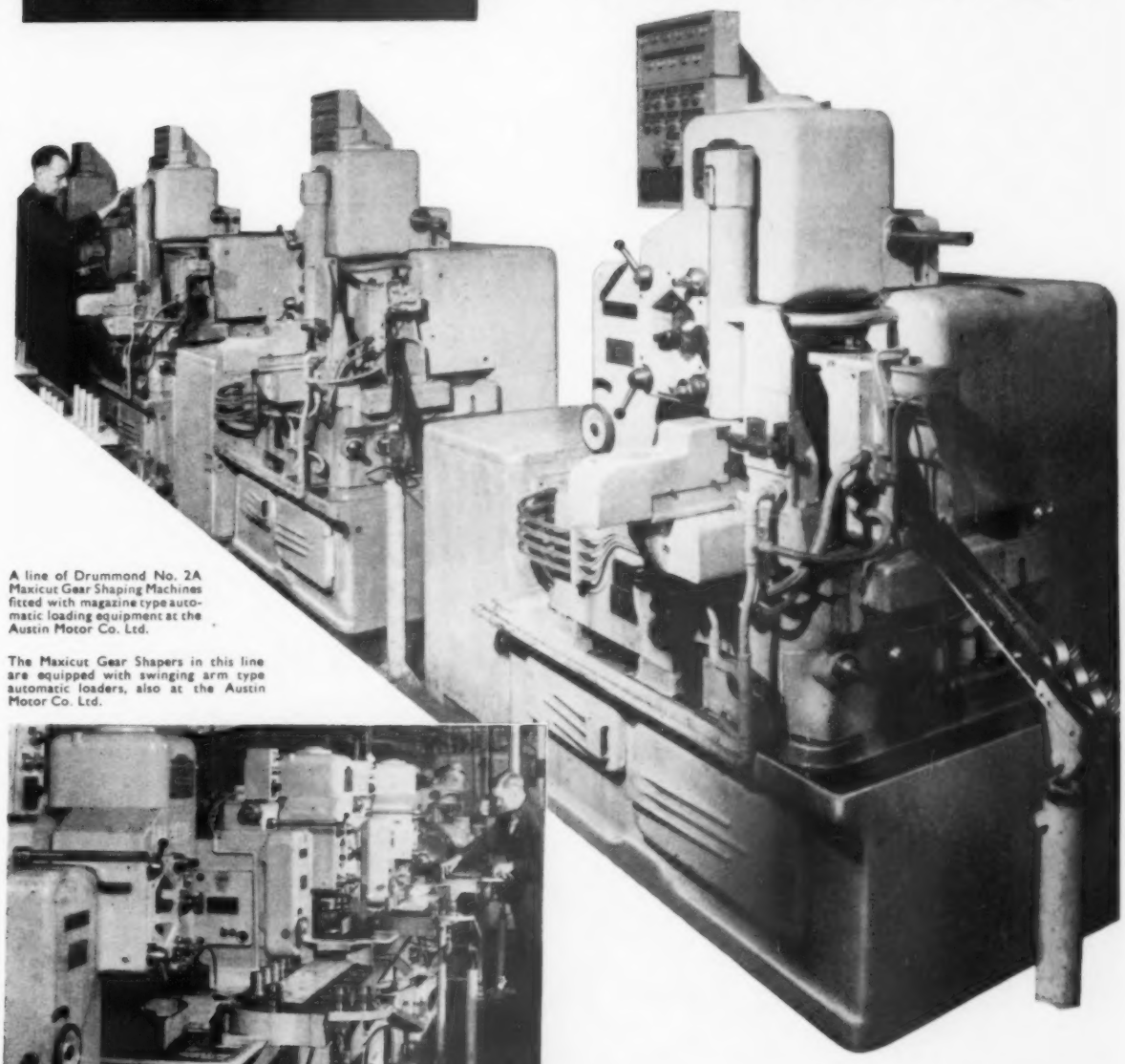

A line of Drummond No. 2A Maxicut Gear Shaping Machines fitted with magazine type automatic loading equipment at the Austin Motor Co. Ltd.

The Maxicut Gear Shapers in this line are equipped with swinging arm type automatic loaders, also at the Austin Motor Co. Ltd.

For the mass production of automobile gears, Maxicut 2A Gear Shaping Machines can be arranged with either magazine or swinging arm type Automatic loading as illustrated. The equipment is generally adaptable for use on a wide range of components for cutting gears, splines etc. Write today for further details.

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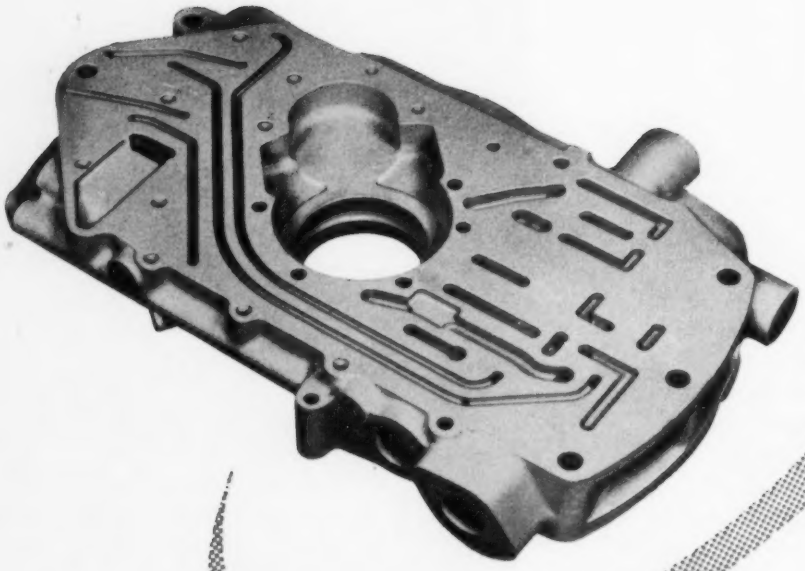

Automatic Transmission Part in Aluminium produced by Shell Moulding. The slots are in the 'as cast' state and are not machined. By courtesy of Hobbs Transmission Limited

STERLING METALS LTD

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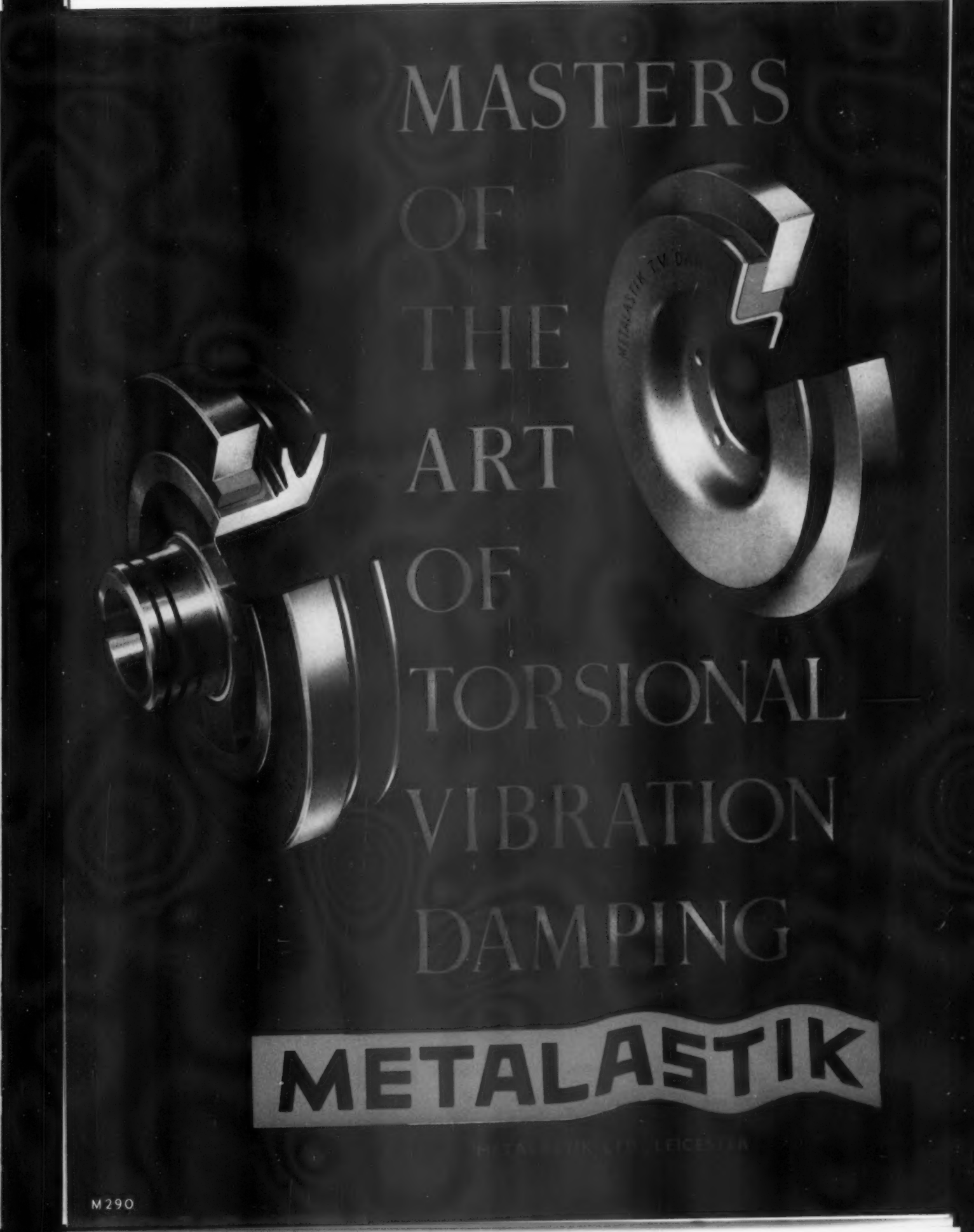
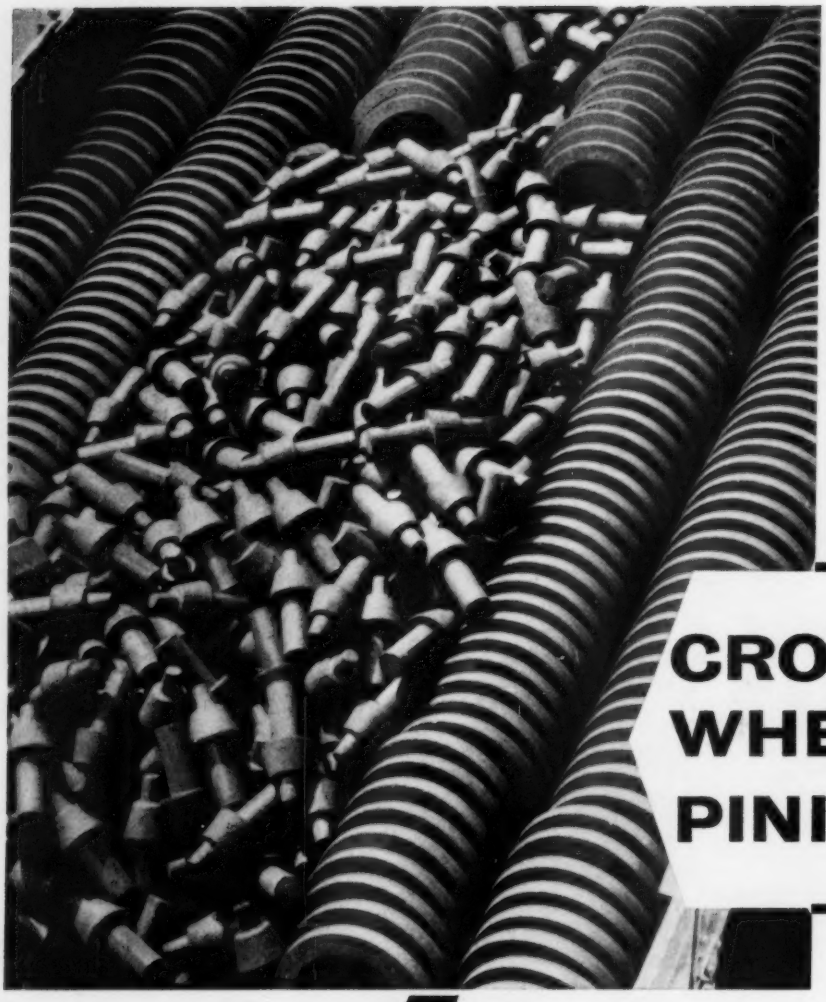


Photo-montage by Geoffrey Gale



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These are manufactured in a range of all normal sizes.
The bucket type (hollow and without windows) has virtually become standard practice and we always recommend this type for new designs where possible. We prefer to supply the fully machined tappet and where the design includes a ball seating, this can be locally hardened without disturbing the chill of the operating face.

THE BRITISH PISTON RING CO. LTD., COVENTRY, ENGLAND

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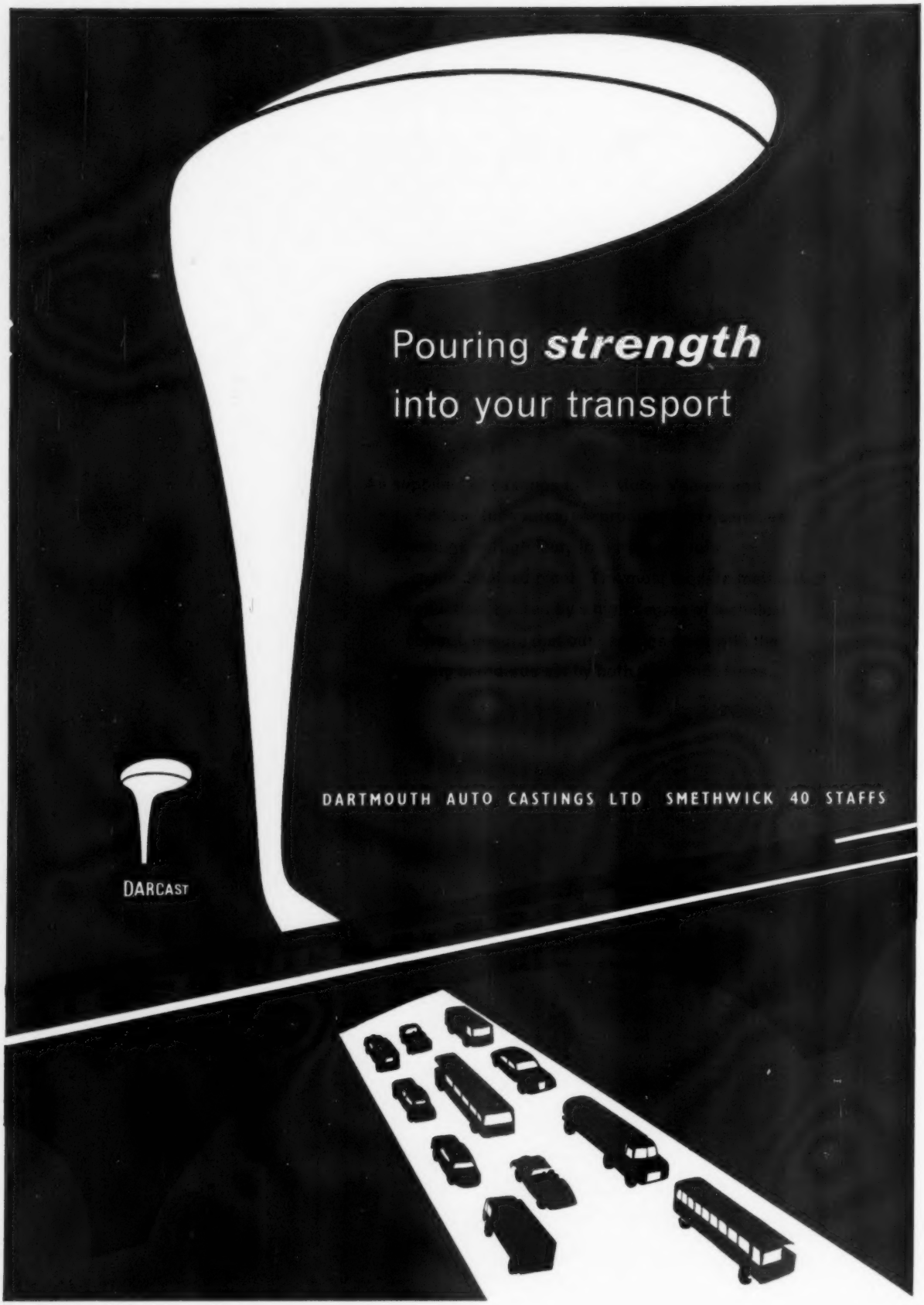
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AUTOMOBILE ENGINEER

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Published the second Wednesday in every month
by ILIFFE PRODUCTION PUBLICATIONS LIMITED
Dorset House, Stamford Street, London, S.E.1
Telephone · Waterloo 3333 (65 lines)
Telegrams · Sliderule, Sedist London
The annual subscription inland and overseas is £3 0s 0d including the special number Canada and U.S.A. $8.50

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Coventry · 8-10 Corporation Street
Telephone · Coventry 25210
Birmingham · King Edward House, New Street, 2
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AUTOMOBILE ENGINEER

DESIGN · MATERIALS · PRODUCTION METHODS · WORKS METHODS

Adhesives for Metal-to-Metal Joints

TWO months ago a new British car was announced in which the bonnet lid stiffening frame is attached to the outer panel by means of an adhesive. Obviously, there are other applications for this method of making metal-to-metal joints. Hitherto, although adhesives have been widely used for this purpose in the aircraft industry, they have been rejected by automobile engineers on the following grounds. The time required to cure the synthetic resin adhesives was too long, elaborate jigs were necessary to apply the requisite pressure to the joints, and the production set-up was further complicated by the fact that heat had to be applied for the curing process.

Now the situation is changed. Synthetic resins are available that cure in about two minutes. While the components are undergoing this process other assembly operations can be made on them, so the curing period does not necessarily have to be idle-time so far as plant utilization is concerned. As is well known, elaborate jigs in any case are to be found in modern production lines for welding operations; in fact, it is probable that the jigs for adhesive jointing would not need to be complicated. With regard to the objection mentioned at the end of the previous paragraph, catalytically cured resins are now available, so the application of heat is unnecessary. Even where heat is desirable, curing might be done in the paint drying ovens.

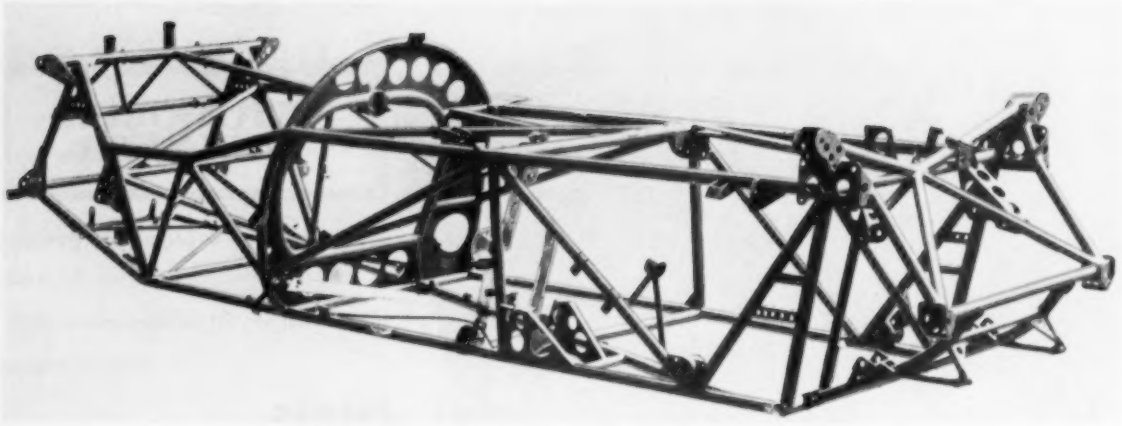
The advantages of adhesives for vehicle body construction are as follows. They distribute the loads more uniformly than is possible with spot welds. Secondly, by virtue of their initial plasticity and their inherent elasticity when cured, they generally obviate any tendency for local stress concentrations to occur during the assembly operation. Moreover, they assist in damping vibrations. Also, synthetic resins are good insulators, so galvanic corrosion at the interfaces should not present any problems. Since the temperatures at which the joints are made, even if heat is required for curing, are much lower than those attained during welding, distortion of the components is obviated. The outer surfaces of the parts joined are unaffected in any way and, therefore, their finish is not impaired for painting. Another feature which, in certain applications, could be useful is the relative ease of making watertight joints.

Synthetic resins used for adhesives make joints having shear and tensile strengths of the order of 10,000 lb/in^2. Fears have been expressed to the effect that stress concentration, due to local peeling-loads at one end, might cause the joint to tear apart. By suitable design, however, this generally can be avoided—it is well known that any glued joints perform best if they are arranged so that they are subjected to only shear loading. Even if this were to be impracticable, the employment of, perhaps, no more than one spot weld at each end would be sufficient to prevent peeling. This latter measure could easily be employed where the layout is such that the end spot welds can be concealed.

The general reliability of bonded joints can hardly be in question. They have for many years been used on aircraft, where structural security is, of course, the prime consideration. Engineers in that field have used bonding for the construction of components of exceptionally complex shape, and of honeycomb structures. Therefore, it should be easily practicable to bond many motor vehicle components.

Currently, of course, the unsightliness of spot welding on exterior panels is accepted for some commercial vehicles, because of the heavy loading to which their structures are subjected. The advent of suitable adhesives provides manufacturers with an opportunity to avoid this expedient. Where, for the sake of external appearance, an alternative and not altogether satisfactory arrangement has been accepted as a compromise, the employment of bonding might give better results and also simplify construction.

Everyone appreciates, of course, that extensive development tests will be necessary before any adhesive can be accepted. Fatigue life of bonded joints under the conditions experienced in road vehicle operation is one of the unknown factors, although in other applications fatigue strength has been proved satisfactory. Not only must the resin stand up to laboratory tests of its physical and chemical properties but also it must be shown to retain those properties when contaminated by hydrocarbon fuels and oils, and under all climatic conditions to which it might be subjected in service throughout the world. Nor must it be forgotten that, in some commercial vehicles, it will be subjected to chemical contamination by the load carried. So far as contamination with hydrocarbons is concerned there should be no difficulty, since there is a large fund of experience in aircraft operation to be drawn upon.



In spite of the large number of tubes necessary for rigidity, the space type frame weighs only 73½ lb. This low weight results from the use of Reynolds 531 alloy steel tubing, mainly of 1 in and ⅞ in diameter; all the joints are Sifbronze welded, thus minimizing the risk of distortion

Ferguson Racing Car

Part II: The Space Type Frame, Body, Front and Rear Suspension, Steering Layout and Braking System

IN accordance with the current practice for racing cars, the Ferguson Project 99 vehicle has a so-called space frame. If correctly designed, a frame of this type—consisting of a multiplicity of small-section tubes—should, of course, be both lighter and torsionally stiffer than one based on two side members of channel, box or tubular section. Admittedly, the stressing problems involved are more complex, manufacture is more difficult and the layout may give rise to some restriction of accessibility, but these disadvantages are clearly of less importance in a racing car than they would be in a vehicle intended for normal use.

The frame of the Ferguson car is constructed of Reynolds 531, 18 s.w.g. steel tubing; all the tubes are of circular section and, with a few minor exceptions, are of either 1 in or ⅞ in diameter. Every joint is Sifbronze welded, because the relatively low temperatures involved minimize the risk of distortion or of embrittlement near the weld. It is noteworthy that the use of high-strength tubing has enabled the weight of the complete structure to be kept down to 73½ lb.

In effect, the structure comprises five bays: the leading one provides anchorages for the radiator and the front final-drive unit and suspension; the engine is installed in the second bay, and the gearbox assembly occupies the third one. Although the driver's seat is in the fourth bay, his legs extend through a bulkhead into the third one, alongside the gearbox. The final bay is the counterpart of the first one, in that it carries the rear final-drive unit and embodies anchorages for the suspension.

Because the disposition of the tubes is clearly shown in the accompanying illustrations, there is no need of a full description, though several points are worthy of special mention. In the first place, the bulkhead mentioned in the previous paragraph, which adds considerably to the torsional stiffness of the structure, is constructed of both tubing and sheet steel. It consists of inner and outer hoops of ⅞ in diameter tubing, which are welded to a 22 s.w.g. sheet steel diaphragm. In the diaphragm are pierced various circular holes, the largest of which is near the bottom left-hand corner: it has a lipped edge, for stiffness, and through it passes the propeller shaft between the gearbox and the rear final-drive unit. In the lowest portion of the diaphragm are four lightening holes, also with lipped edges, and at the top are some unlipped holes to accommodate the Smiths instruments.

The 1 in diameter top rails of the frame are continuous from the beginning of the second bay to the end of the fourth bay. Owing to the offset of the driver to the right, the right-hand rail has a dip in the fourth bay, to facilitate access, and to give adequate shoulder room. The bottom rails are ⅞ in diameter and are continuous from the beginning of the second bay to the rear end of the frame. Lozenging in the horizontal plane is resisted mainly by a single diagonal member at the bottom of the frame, between the right-hand front corner of bay 1 and the left-hand rear corner of bay 4. This member is continuous, and it interrupts the two intermediate cross tubes. Additional resistance to lozenging is afforded by two tubes at the top of the frame: one is a diagonal in bay 3, from the left-hand front corner to the right-hand rear corner, and the other extends from the left-hand rear corner of bay 2 to the middle of the cross tube at the front of this bay.

In Part I of this article, reference was made to the asymmetry of the engine mounting arrangements, on account of the compound angle of installation. Because of the obliquity of the unit as viewed in plan, the housings for the Silentbloc bushes of the front mountings are attached to the frame at slightly different positions. The right-hand housing is situated in the angle between two inclined triangulating tubes, immediately above the bottom rail, whereas the left-hand one is slightly further forward, in the angle between the other bottom rail and one of the corresponding triangulating tubes. Gusset plates reinforce both joints. Although the lower mountings of both the final-drive units are of common design, the anchorage bolts being

carried in brackets welded to the frame tubes, their upper mountings are different: the fabricated brackets for the bolts of the front unit are integral with the frame, but those for the rear ones are bolted on.

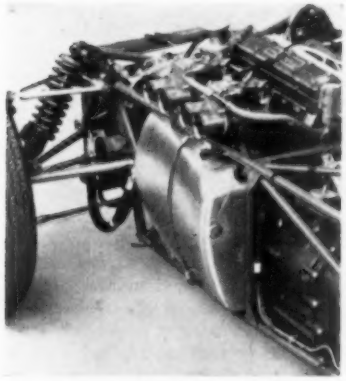
The anchorages for the upper ends of the suspension units comprise fabricated brackets of channel section welded to the four top corners of the frame. Immediately below these brackets are seatings for the bolted-on pivot brackets for the upper wishbones. At the rear, both pivot brackets for the lower wishbones are integral with the main structure, but at the front, only the foremost brackets are integral, the trailing pair being bolted on to facilitate assembly.

Under the current Formula 1 regulations, all cars must be provided with a roll-over hoop. This has to be mounted behind the driver and is supposed to be sufficiently robust to withstand the weight of the car if it should overturn, thus reducing the risk of serious head injury to the driver. In the case of the Ferguson car, the hoop is of 20 s.w.g. material and is detachable, to facilitate the mounting or removal of the bodywork; its ends spigot into two sockets, welded to the upper transverse tube between bays 4 and 5, and is secured by bolts that pass through holes drilled in the tubes and the hoop.

Body construction

The styling of the handsome, low-drag body was the responsibility of Walter Belgrove, formerly Chief Body Engineer of the Standard Motor Co, who acted as consultant to Harry Ferguson Research Ltd. throughout the design and mock-up stages. Williams and Pritchard Ltd. built the shell, which comprises eighteen hand-beaten components of 18 and 20 s.w.g. aluminium. These are the under-tray, which is in three parts, the four-piece nose assembly, the tail, the top cowling over the engine, two panels on each side, the cockpit cover, and four small panels round the upper portions of the suspension. Most of the panels are attached to each other and to the frame by means of Dzus fasteners. An exception, though, is the top cowling of the engine: since it has to be quickly detachable, this panel is secured by two spring-loaded clips on each side, mounted on the leading pair of side panels. In the panel are bulges to clear the carburettors and the camshaft housings.

On each side of the domed tail section is a large indentation, shaped to follow the body profile; these indentations form outlets for the air heated by the brakes, which are enclosed by the shell. There are similar louvres in the leading side panels, to assist the cooling of the front brakes. The exhaust pipes emerge through the right-hand panel.



The oil tank, secured by an elastic strap, is installed beside the engine, on the left of the car. It is of aluminium sheet and, though having a capacity of 5 gal, it only contains 2 gal, to allow for frothing

Because of the offset driving position, the cockpit cover is asymmetrically mounted. On the left, its lower edge is level with the upper wishbones of the rear suspension, but the corresponding right-hand edge is at approximately the level of the lower wishbones. The opening in the cover is also displaced to the right, and its edge is lipped inward for stiffness. A Perspex windscreen, moulded by Thermoplastics Ltd. and of the now-popular horseshoe form, is secured to the cover by eight screws and nuts. The cockpit cover is internally reinforced at each side, just ahead of the windscreen, to carry the two rear-view mirrors. To keep the overall height as low as possible, the backrest of the Microcell lightweight seat has a considerable rake.

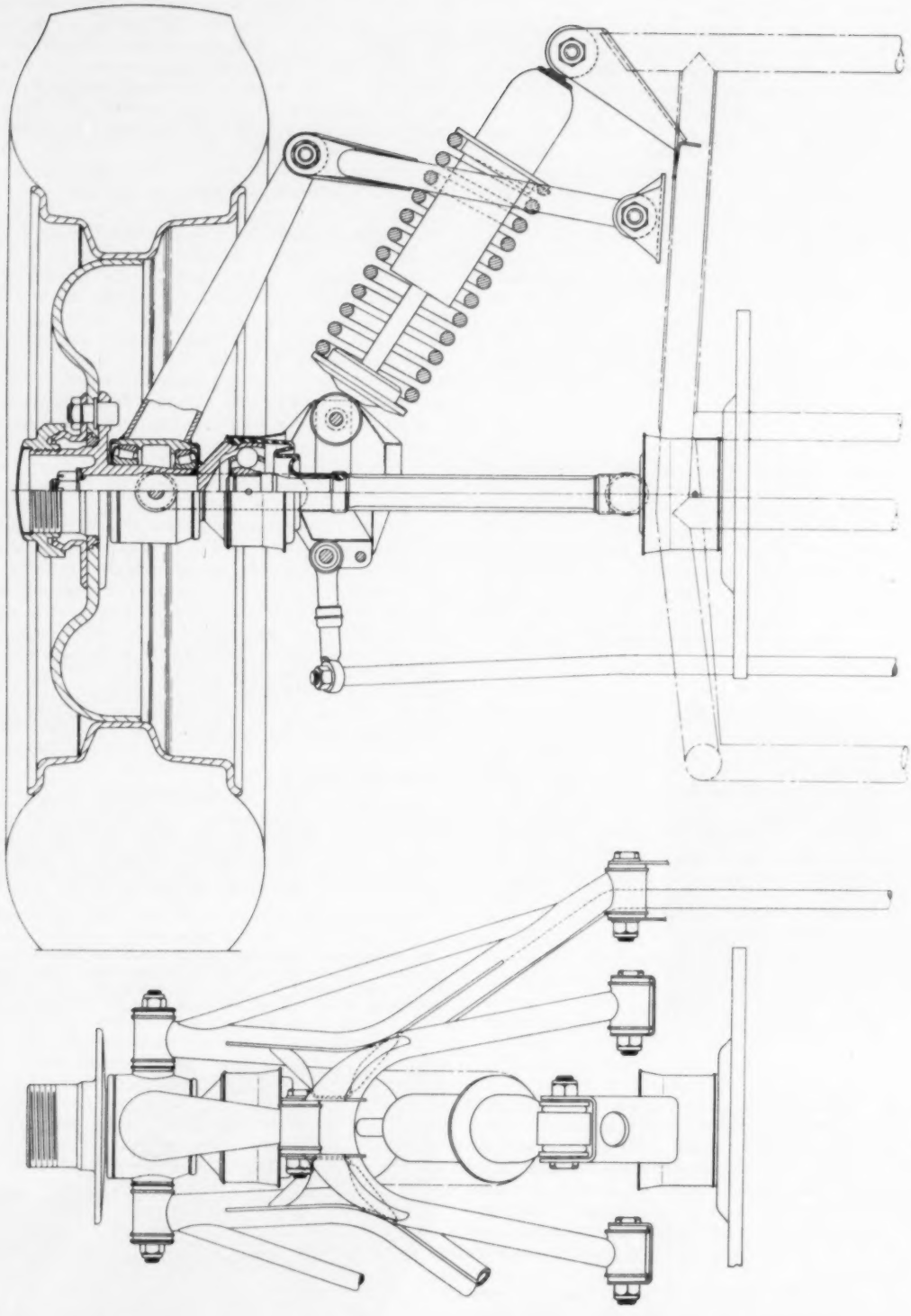
Enclosed within the body shell are the two fuel tanks. The larger of these, a Williams and Pritchard unit of aluminium and with a capacity of 14½ gal, is at the rear. Rubber is interposed between it and the frame, to which it is secured by two elastic straps. Its filler cap is accessible through a hole in the top of the tail cowling. The second tanks has a capacity of 10½ gal and is of the flexible type, made by Marston Excelsior Ltd. It is mounted on the left side of the vehicle, alongside the driver, in the space between the frame and the side panelling. By dividing the capacity in this way, and drawing from both tanks simultaneously, variations in the handling characteristics with fuel load are kept within reasonable limits. In fact, the variation in rear wheel loading from the full to the empty condition is only 8 per cent.

The aluminium oil tank—another Williams and Pritchard product — has a capacity of 5 gal though it contains only 2 gal, the surplus being to accommodate frothing. It is also mounted on the left, alongside the engine, and fits between the top and bottom rails of the frame. Location is effected by one of the diagonal tubes, which registers in a channel formed in the inboard face of the tank; an elastic strap forms the securing means. The filler neck projects inward from the leading end of the tank, and the cap is below and immediately ahead of the front carburettor, for accessibility when the engine cover is removed.

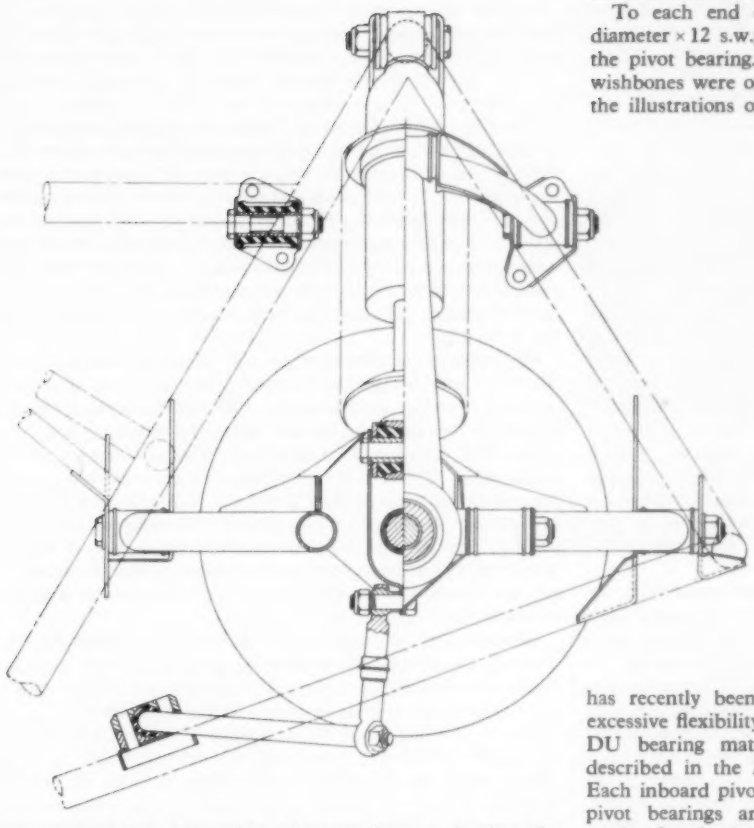
Suspension systems

In order to give the desired almost neutral steering characteristics, having regard to the balanced driving torque and weight distribution, the front and rear suspension systems have a similar layout, consisting of double transverse wishbones diverging towards the wheel. This system

This three-quarter rear view clearly shows the clean lines of the aluminium body, which consists of 18 components. The indentations in the sides, behind the suspension wishbones, are to assist the ventilation of the Dunlop disc brakes, which are mounted inboard



GENERAL ARRANGEMENT OF THE REAR SUSPENSION OF THE FERGUSON PROJECT 99 RACING CAR



Since the lower wishbone is at the same height as the half-shaft, no variation in the effective length of the shaft occurs when the suspension articulates. Glacier DU bearings, instead of the rubber units illustrated, are now used for the pivots of the wishbones. The Dunlop light alloy wheels are quickly detachable; and pegs transmit the drive from the hubs

was preferred to the swing axle type because it gives the designer a wide choice of roll centre heights and has more favourable characteristics in respect of changes of track and camber angle with wheel displacement. The layout, which was evolved by Harry Ferguson Research Ltd, is the subject of patents because it departs in several ways from conventional practice. Of these unusual features, the most obvious is the high position of the lower wishbones, which are level with the half-shafts. An important advantage of this disposition is that articulation of the suspension does not result in any length variation of the shafts. In addition, since the lower wishbones react virtually the whole of the longitudinal components of the dynamic loads, the duties of the upper wisbones are much less arduous than in an orthodox layout.

Unsprung weight of the suspension systems has been minimized by the inboard mounting of the brakes, the reactions of which are, of course isolated from the suspension. An anti-roll bar is installed at each end, to enable relatively low-rate springs to be used—in the interest of wheel adhesion and riding comfort—without making the roll stiffness undesirably low. Combined coil spring and damper units comprise the suspension medium.

For clarity, the rear suspension will be studied in detail first, after which the differences of the front suspension will be mentioned. In the rear system, each arm of the lower wishbones consists of a main and a secondary tube, both of Reynolds 531 material. The dimensions of these tubes are 1 in diameter × 16 s.w.g. and ½ in diameter × 18 s.w.g. respectively. Both main tubes of one wishbone are bent, just outboard of the middle, and are disposed so that—as viewed in plan—they are closest together at the bends; the outboard convergent angle is much smaller than the other. Each secondary tube is scarfed at the ends and welded to the main tube to reinforce it across the bend.

To each end of both main tubes is welded a 1⅛ in diameter × 12 s.w.g. tubular eye, which forms a housing for the pivot bearing. Originally, all the pivot bearings of the wishbones were of the unbonded, rubber type—as shown in the illustrations of the general arrangements—but a change has recently been made to Glacier DU bushes, to avoid excessive flexibility and variable friction in the system. The DU bearing material, which embodies p.t.f.e, was fully described in the May 1960 issue of *Automobile Engineer*. Each inboard pivot has its individual bolt, and the outboard pivot bearings are mounted on studs integral with the carrier for the hub bearings.

The half-shaft lies between the arms of the wishbone but, for constructional reasons, the geometric centre of the outboard universal joint is situated appreciably inboard of the adjacent pivot axis of the wishbone. As a result of this lateral offset, the half-shaft and the wishbone do not articulate as one. The centre of the inboard universal joint is therefore also displaced inboard of the other pivot axis of of the wishbone; however, this second offset is slightly smaller than the first, to keep the length of the half-shaft virtually constant in spite of the variations of camber angle inevitable with any unequal-length wishbone system.

Between the bends of the two arms of each lower wishbone is a bridge structure comprising two fabricated brackets and three bolts. One of the brackets is welded to each main tube, and their lower ends are clamped together, below the half-shaft, by two bolts, one of which also secures a link that projects downward to the end of the anti-roll bar. Connecting the upper ends of the brackets is the third of the bolts just mentioned, on which is carried the lower end of the spring and damper unit; the inner sleeve of the rubber bearing in the eye of the unit forms a spacer between the brackets.

Each upper wishbone consists basically of a single tube, ¾ in diameter × 16 s.w.g, again of Reynolds 531 material, bent into the form of a V. So far as loading is concerned, a simple link would have sufficed, but it was necessary to provide clearance over the spring and damper unit. To the ends of the arms of the V are welded tubular eyes that house the pivot bearings. In the earlier arrangement, shown in the accompanying half-tone illustration, the pivot bolts of

these bearings were screwed into bosses at the ends of a single carrier bracket, which was of fabricated construction and was bolted to the frame at four points. As can be seen from the general arrangement, however, a separate bracket is now employed for each bolt. The design of these brackets can be varied to alter the pivot position, and hence the roll centre height, and shims can be fitted between them and the frame to modify the camber angle.

Projecting transversely from the apex of the V are two small, parallel plates carrying the outboard pivot bolt. Gussets reinforce the joints between the plates and the tube. The Glacier DU pivot bearing is housed in a tubular eye at the upper end of a king-post, the lower end of which is welded to the hub bearing carrier mentioned earlier. This king-post, which is machined from solid En.36 material, has the form of an elongated and truncated cone, and its diameter decreases from 2 in at the bottom to $\frac{7}{8}$ in at the top.

Details of the opposed, taper roller hub bearings were given in Part I of this article. The bearing housing is also of En.36 steel, and in its bore are machined two integral collars forming abutments for the outer races. Each bearing is sealed by a Nilos ring of the type described in the March 1956 issue of *Automobile Engineer*. In addition, there is a pressed steel dust cap at each side of the housing: that at the outboard side is secured between the hub flange and the inner race of one bearing, and the other is pressed on to the driven member of the universal joint. Both caps therefore have running clearance over the periphery of the housing.

In the reproduced drawing of the rear suspension, an anti-roll bar of $\frac{1}{2}$ in diameter is shown, but others have been tried, to vary the handling characteristics. As already mentioned, the ends of the bar are connected by links to the bridge structure of the lower wishbones. The transverse portion is carried in split type Duralumin bearings mounted on brackets welded to the bottom rails of the frame; this design of bearing was found preferable to the rubber bush illustrated. It is worthy of mention that the $\frac{1}{2}$ in diameter bar increases the basic roll stiffness at the rear by 33 lb-ft/deg.

The telescopic dampers, which are of Armstrong manufacture, have a bore of $1\frac{1}{8}$ in and incorporate the bump and rebound stops. For the initial trials, use was made of units embodying a manual adjuster for the setting of the rebound valves. Once the desired characteristics had been determined, however, the adjustable units were replaced by the lighter and non-adjustable GT7 type.

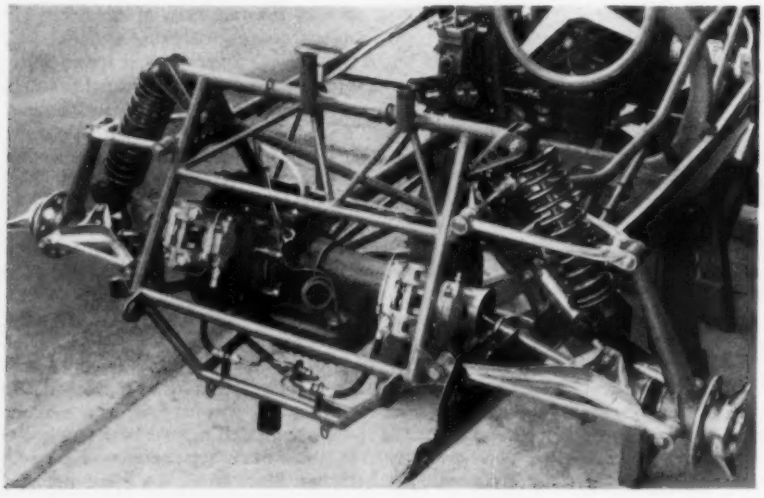
As measured at the wheel, the suspension movement between the stops is 6 in, equally divided between bump and rebound. With considerable assistance from The Tempered Spring Co. Ltd, experiments have been carried out with springs of various dimensions and rates: those employed at the time of writing have a wire diameter of 0·420 in, an outside diameter of 3·19 in, and nine effective coils. The rate at the static load position, again measured at the wheel, is 111 lb/in when the anti-roll bar is disconnected.

Other main dimensions of the rear suspension system are as follows: effective lengths of lower and upper wishbones, $12\frac{1}{16}$ in and 9 in respectively; pivot bases of lower and upper wishbones, 13 in and $6\frac{3}{4}$ in respectively; basic camber angle at static load position, nil; maximum variation in camber angle, -2 deg on bump and $+3$ deg on rebound; maximum variation in track, $+0·6$ in on bump and $-1·6$ in on rebound; periodicity, 120 c/min; roll centre height, 5·5 in at static load position; roll stiffness, with anti-roll bar connected, 139 lb-ft/deg.

The differences between the rear suspension and that at the front are dictated primarily by the necessity of incorporating steering arrangements. In the front suspension, therefore, ball joints replace the outboard pivots of both the upper and the lower wishbones. The two lower joints enclose the outboard universal joints of the half-shafts, and are concentric with them. Since the inboard pivot axis of each lower wishbone passes through the geometric centre of the inboard universal joint, the half-shaft and lower wishbone articulate as one, so there is no inherent tendency for the length of the half-shaft to vary with suspension movement.

Whereas both the upper and the lower rear wishbones are symmetrical about their transverse axes, as viewed in plan, the front wishbones are not. This asymmetry is due partly to the need of providing castor angle, and partly to the layout of the front end of the frame. The pivot bases for the upper and lower wishbones are respectively 7 in and $12\frac{1}{2}$ in, while the effective lengths of the two are $9\frac{1}{8}$ in and 13·18 in.

Because the anchorage of the lower end of the suspension unit to the lower wishbone is different from that at the rear, the arms of the wishbone are single, straight tubes. The upper wishbone, too, is of different design from its counterpart at the rear: each arm is a separate, straight tube, the outboard ends being brazed into lugs that extend from the ball joint housing. Both the material specification and the tube dimensions of the wishbones are the same as at



The pivot axes of the lower wishbones are situated outboard of the centres of the universal joints although, at the static load position, they lie in the same horizontal plane. Modifications made to the layout since it was photographed include the adoption of a separate bracket for each of the two inboard pivot bolts of the upper wishbones, and of non-adjustable dampers. Octagonal wheel-retaining nuts have replaced the eared type illustrated

the rear — except that the front lower wishbones are of 10 s.w.g. CDS10 tubing—and the inboard pivot bearings, too, are similar, as are the inboard pivot brackets for the upper wishbones.

The upper ball joint is of orthodox design, with a single-diameter ball and a two-piece housing. It is adjustable for wear by means of shims, which are inserted between the main, lower portion of the En.36 housing and the screwed-in upper portion. A very neat and robust mounting has been adopted for the ball pin. This component has a serrated collar under the head, and a relatively long shank; the lower portion of the shank is of smaller diameter than the upper portion and is threaded. The pin is screwed into a boss at the top of the king-post, which is of similar design to that of the rear suspension, and the upper portion of its shank forms a spigot in the hole in the boss. After the pin is tightened, by means of a special spanner that fits the serrations, it is locked by drilling a hole through the boss and the shank, and driving in a pin.

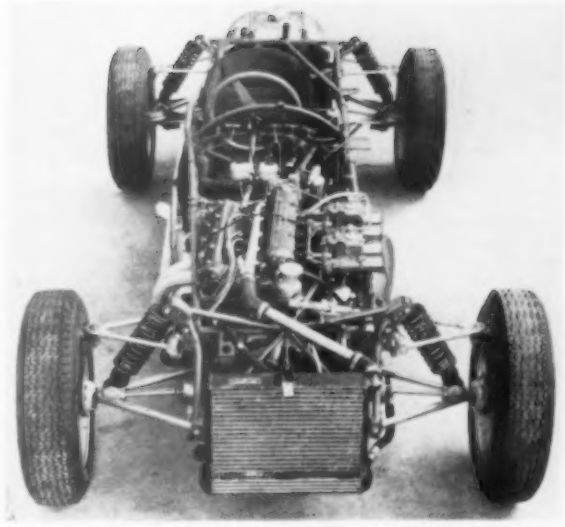
Each lower ball joint has three main components, the two-piece outer member and the inner member. The outboard hemisphere of the outer member is of En.36, and it has a cylindrical extension that houses the hub bearings, in the manner already described in connection with the rear suspension. To its upper side is welded the king-post, and underneath is a facing to which is bolted the steering arm. On the joint face is a shallow counterbore, in which is located a spigot machined on the other portion of the outer member. A part-spherical, phosphor-bronze bearing ring is riveted to the inside of the outboard hemisphere.

Subsequent to the preparation of the drawing that formed the basis of our illustration of the general arrangement of the front suspension, changes have been made to the design of the inboard hemisphere. Originally, this component was in one piece, also of En.36 steel, and to its internal surface was riveted a second phosphor-bronze bearing ring. To facilitate assembly, though, a split component is now employed: it is of Duralumin and bears directly on the inner member of the joint. The split is in the horizontal plane through the transverse axis, and the two halves are retained and located by a steel keeper ring, which has a conical inner face that seats on a mating external face on the half-bearings. Eight $\frac{1}{4}$ in socket-head set-screws hold the components of the outer member together: they pass through holes in the flanges of the ring and the half-bearings, and screw into that of the outboard hemisphere.

The inner member of the joint is of En.36 material and, from its spherical portion, a cylindrical extension projects in an inboard direction. To this projection are welded the arms of the lower wishbone and the bracket to which the lower end of the suspension unit is attached. The working surface of the inner member is Parco-Lubrized to minimize the risk of scuffing.

Use is made of the space between the two bearing surfaces of the outer member as a reservoir for the grease with which the joint is packed on assembly. To prevent any foreign matter from finding its way into the universal joint, the half-shaft carries a shroud of approximately hemispherical form, which is situated within the ball joint. This shroud overlaps the driven member of the universal joint, on the periphery of which is mounted an annulus carrying a Prescollan polyurethane rubber sealing ring of square section, bearing lightly on the interior of the shroud.

The springs of the front suspension differ from those of the rear suspension in that the wire diameter is 0·40 in, the outside diameter is 3·15 in and there are $9\frac{1}{2}$ effective coils. With the anti-roll bar disconnected, the rate at the wheel is 90 lb/in. At the static load position there is basically a zero camber angle; the maximum variation of this angle is $-3\frac{1}{2}$ deg on bump and $+1\frac{1}{4}$ deg on rebound, whereas that of the track is $+0{\cdot}9$ in on bump and $-1{\cdot}4$ in on rebound The periodicity is 110 c/min.



In the case of the front suspension, the pivot axes of each lower wishbone pass through the geometric centres of the universal joints. The curved, fabricated king-posts shown here have been superseded by conical components machined from the solid; they are welded to the hub bearing housing, which also forms part of the lower ball joint

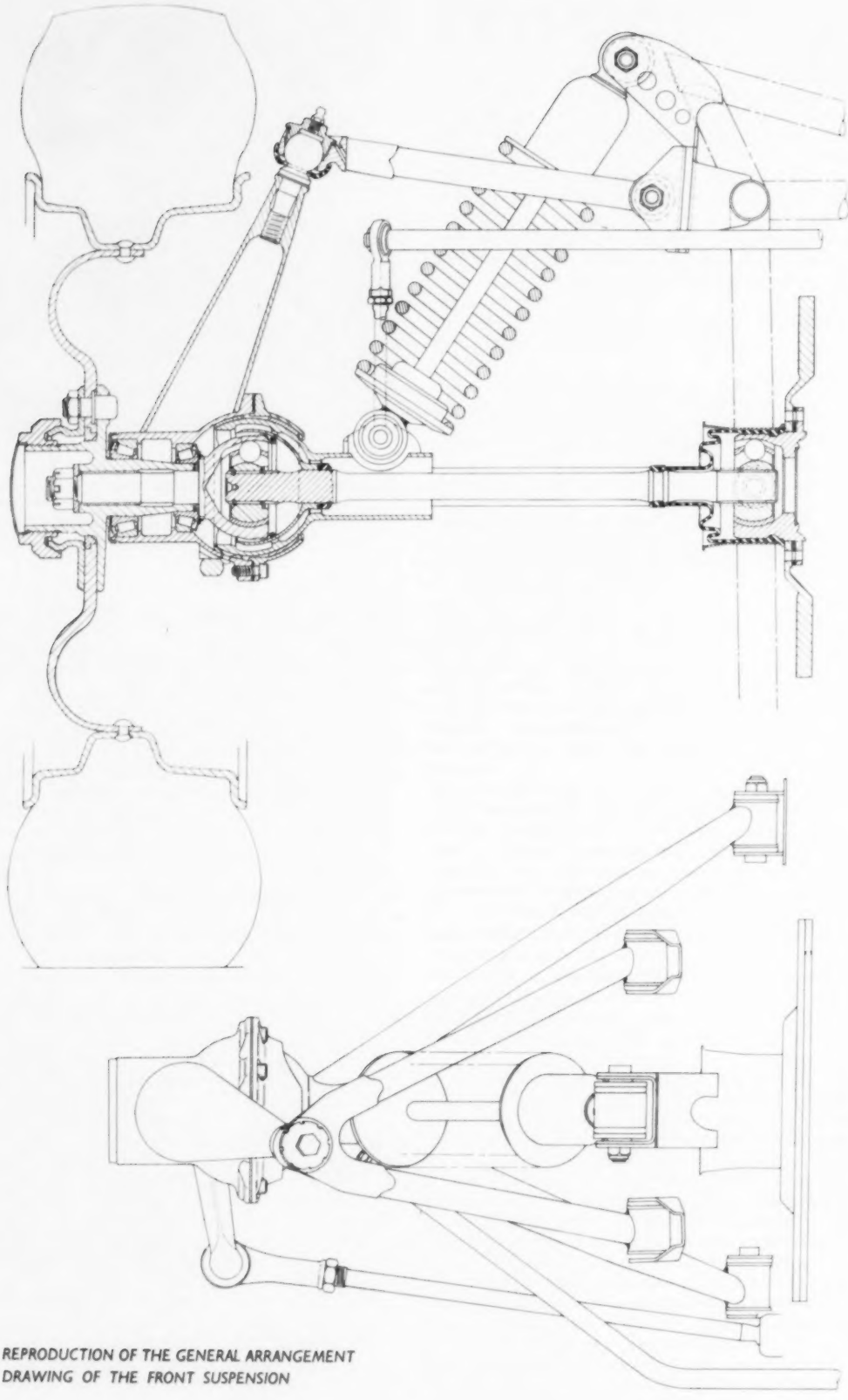
It will be appreciated that, for the lower ball joint and universal joint to be concentric, the outboard pivot of the lower wishbone—that is, the ball joint—must be further inboard of the centre plane of the wheel than in the case of the rear suspension. Consequently, to provide the same basic roll stiffness at both ends, the front track has been made 2 in wider than the rear track, thus equalizing the transverse distance between the wishbone ends in both systems. Owing to the difference in the geometry, the roll centre height of 5 in is slightly less than at the rear.

Although the front anti-roll bar, as shown in the general arrangement, has the same diameter as its counterpart at the rear, its installation is different. The connecting links at its ends project upward instead of downward from the lower wishbones, to which they are attached adjacent to the anchorages of the spring and damper units. As a result, the bearings carrying the transverse portion of the bar are mounted on the underside of the top rails of the frame; they are of the same type as those of the rear anti-roll bar.

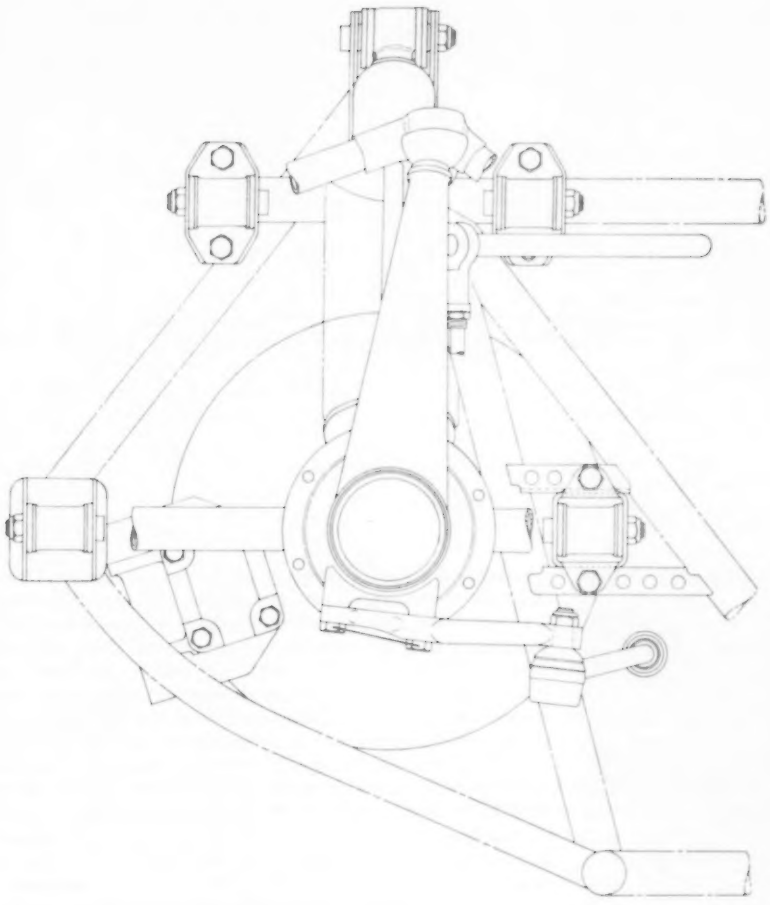
Steering layout

A rack and pinion steering mechanism has been adopted for the Ferguson Project 99 car, but the layout is rather more complex than is usual with this layout. The complexity arises from the fact that the oblique engine installation necessitates a considerable offset of the pinion spindle to the right of the car's longitudinal axis, as viewed in plan. If the tie rods to the steering arms were attached directly to the ends of the rack, it would be necessary for the right-hand one to be shorter than the optimum length dictated by the suspension geometry, with the consequent likelihood of excessive kick-back on bumps.

To obtain a geometrically accurate system, therefore, the tie rods are connected to an intermediate or slave rod; this is rigidly attached to the left-hand end of the rack, and is guided to maintain parallelism. This layout, which is shown in one of the accompanying illustrations, has the secondary advantage of enabling the rack assembly to be mounted



REPRODUCTION OF THE GENERAL ARRANGEMENT DRAWING OF THE FRONT SUSPENSION



A very neat and robust arrangement has been adopted for the mounting of the ball pin at the top of the king-post. This illustration shows the original design of lower ball joint, in which a bearing ring was riveted within each half of the outer member. In current joints, the inboard half is Duralumin and bears directly on the inner member; it is split horizontally, to facilitate assembly, and a keeper ring is used to ensure accurate location of the halves

lower on the frame than would otherwise have been possible, thus bringing it clear of the magneto. A three-piece steering column, embodying two Hardy Spicer universal joints of Hookes type, is employed to bring the wheel into the desired position in relation to the driver.

The rack and pinion unit was designed and manufactured by Harry Ferguson Research Ltd. Its light alloy casing is secured by two fabricated brackets to the frame transverse tube that lies below the front of the engine. To avoid excessive reversibility of the gearing, the pinion has a helix angle of 34 deg. Since the axes of the rack member and the pinion make an angle of 86 deg, to provide the necessary offset of the leading portion of the column, the rack teeth are inclined at 30 deg to the axis.

There are six teeth on the pinion, which is of En.39A steel and is integral with its spindle. The spindle is carried in two cup-and-cone ball bearings, adjustment of which is effected by means of shims. Its rear end is serrated to take the leading portion of the column, which is secured by a combined pinch bolt and cotter. In the conventional manner, the rack is pre-loaded against the pinion by a spring-loaded pad; the thrust of the spring is adjustable.

The extended left-hand end of the rack is shouldered and threaded to form a mounting for the fabricated bracket carrying the slave rod. Both the construction of this bracket and the method of its attachment to the rack are clearly visible in the illustration. The sleeve portion of the bracket is a push fit on to the slave rod and is secured by two pins, which are driven into diametral holes drilled through both the sleeve and the rod.

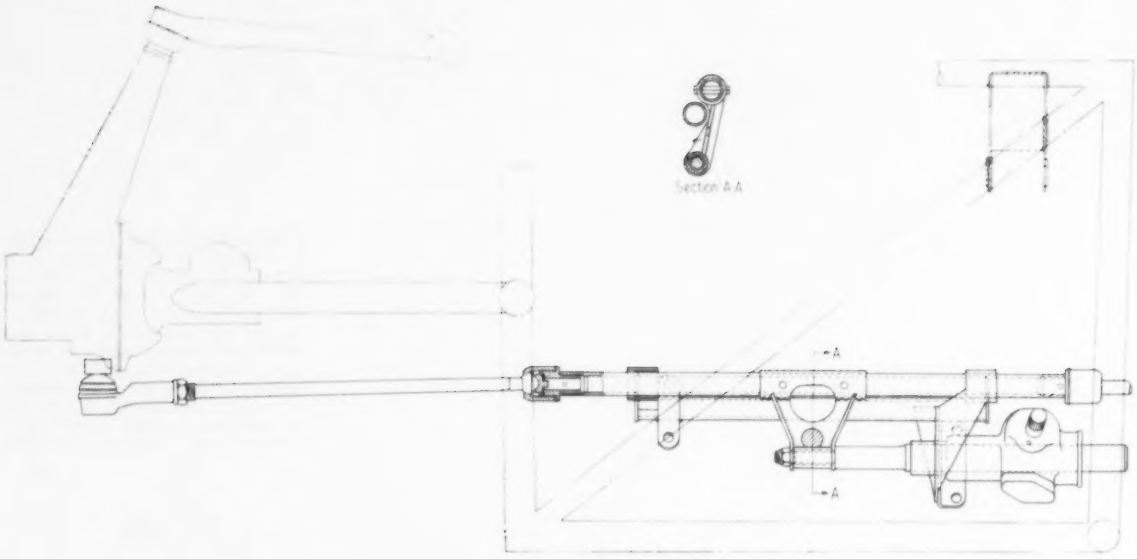
Two Glacier DU bushes, situated $11\frac{1}{4}$ in apart, form the guides for the slave rod, which is, in fact, a tube of $\frac{3}{4}$ in diameter and 12 s.w.g. These bushes are housed in a long, transverse bracket attached to the frame and located on the rack casing. The tie rods of the track rod assembly are solid and of $\frac{7}{16}$ in diameter; they are connected to the slave rod by ball joints of the double-diameter type. No spring loading is embodied in these joints, any adjustment for wear being effected by shimming.

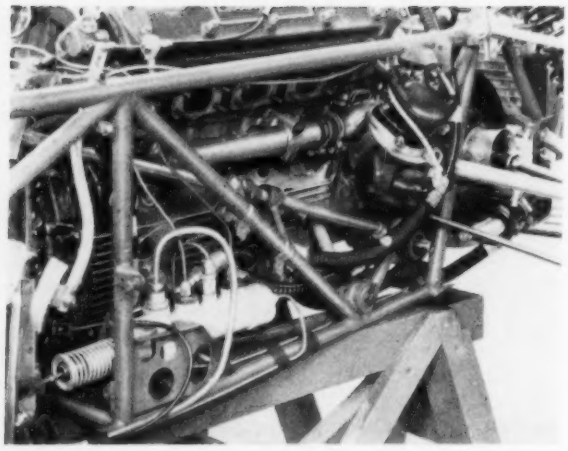
The effective distance between these ball joints is $18\frac{7}{16}$ in and, in the straight-ahead position of the steering, each is just inboard of the pivot axis of the lower wishbone, which is about $2\frac{7}{8}$ in above it. Screwed on to the outboard ends of the tie rods are end-pieces of conventional design, connected to the steering arms by two additional ball joints. The nominal effective length of the tie rods is $14\frac{1}{4}$ in. As viewed in plan, the steering arms are parallel to the centre planes of the wheels, but an Ackermann effect results from the fact that the tie rods trail slightly from the arms to the slave rod.

A castor angle of 6 deg is provided by the rearward inclination of the king-posts; this angle is equivalent to 1·35 in of trail with the standard 5·00—15 in tyres. The swivel axis is inclined laterally at $8\frac{1}{2}$ deg, and it meets the road surface 1·08 in inboard of the geometric centre of the tyre contact patch. No toe-in is given to the wheels.

Wheels, tyres and brakes

Both the front and the rear wheels, which are of Dunlop manufacture, have 15 in diameter, 5 K rims with a width of 5 in. They are fitted with Dunlop R5P tyres; in certain conditions, 6·00 section tyres may be fitted to one or both pairs of wheels instead of the 5·00 in covers mentioned in the

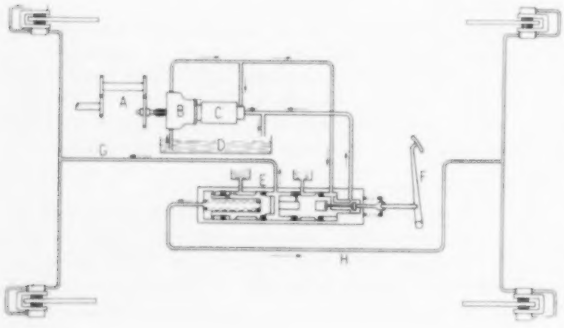




The rack and pinion steering unit is mounted immediately ahead of the engine, the position of which made it necessary to embody two universal joints—one of which is seen here—in the steering column. A tandem type master cylinder is used in the hydraulic brake system

Diagrammatic representation of the unusual Ferguson braking system

A *drive from gearbox;* **B** *Hobourn-Eaton pump;* **C** *Dunlop Maxaret unit;* **D** *gearbox oil sump;* **E** *tandem master cylinder;* **F** *brake pedal;* **G** *pipeline to front brakes;* **H** *pipeline to rear brakes*



previous paragraph. Both the rims and the pierced nave portions of the wheels are aluminium alloy pressings, and are riveted together; the wheels are quickly detachable, being secured by a special octagonal nut on the end of the spindle. The torque is transmitted from the hub to the wheel by pegs, instead of by the splines commonly employed on wire wheels of this type. There are five of these pegs on the wheel, and they register in holes on the hub flange; part-conical seating faces on the nut and the wheel ensure centralization. This ingenious construction is clearly shown in the line illustrations of the suspension systems.

The 10 in diameter discs of the Dunlop hydraulic braking system are mounted inboard for two reasons, the first of which is, of course, the already mentioned one of minimizing the unsprung weight. In addition, this arrangement enables the king-posts to be tucked further into the wheels than would be possible with outboard brakes, thereby permitting a more compact and robust layout. As was explained in Part I of this article, the discs and the driving members of the inboard universal joints of the half-shafts are bolted to adaptors mounted on the output shafts of the final-drive units.

To reduce the transmission of heat from the brakes to the final-drive units, a Ferobestos insulating washer is interposed between the attachment flange of each disc and the face to which it is bolted. Moreover, the heat path between the working portion of the disc, which is $\frac{1}{4}$ in thick, and the flange is restricted by reducing the thickness of the intervening portion to $\frac{1}{32}$ in. To maintain adequate rigidity, the thinner portion is dished. Machined at each end of each final-drive casing are facings to which the caliper assemblies are bolted. At the left-hand end there are actually two facings, disposed mutually at 180 deg; this is because the end assembly is common to both the front and the rear unit, but is mounted on each in a different position, as was explained in Part I. On this account, the calipers of the front and rear brakes are respectively ahead of and behind the transverse axis of the final-drive unit concerned.

Caliper assemblies of the standard, light-weight racing type have been adopted. Each carries two segmental pads of Ferodo DS11 material, having an area of 2·8 in^2 and a thickness of $\frac{3}{8}$ in. The total swept area of the brakes is 352 in^2, equivalent to 573 in^2/ton of the starting line weight of the vehicle plus driver. Although 68 per cent of the

To enable the steering tie rods to be of the right length relative to the geometry of the suspension, they are attached to a slave rod mounted upon the rack instead of being connected directly to the ends of the rack

braking effort is applied to the front discs, the proportioning of that effort is less critical than on an orthodox vehicle, owing to the compensating effect of the transmission control unit. It will be appreciated, however, that a basically incorrect distribution of the effort would increase the work done by the transmission.

Although this unit prevents the wheels at one end from revolving substantially faster or slower than those at the other end, it cannot of itself prevent all four wheels from locking if excessive braking effort is applied. For this reason, a single Dunlop Maxaret anti-skid device is incorporated in the braking system. It is, of course, one of the advantages of the Ferguson transmission system that only the one device is needed, whereas on an orthodox vehicle one is required for each wheel.

The Maxaret unit is mounted to the rear of the secondary casing of the gearbox. Its input spindle is connected by a muff coupling to the spindle of a gear that meshes with the rearmost of the three on the sleeve member driving the clutches of the control device. It is, therefore, subjected to the speed variations of the transmission system as a whole, and comes into action if this is caused to decelerate too rapidly.

Normally, Maxaret units are only used in conjunction with a fully pressurized hydraulic circuit, the pressure being generated by a pump. On the earlier prototypes of the Ferguson passenger car, a different layout is employed, in which the control valve of the Maxaret acts directly on a vacuum servo, embodied in a conventional, unpressurized hydraulic system. For the racing car, however, it was decided to change to a hydraulic servo, which is lighter and more compact, but this still acts on a static hydraulic system of orthodox type.

The hydraulic boost is provided by a Hobourn-Eaton roller type pump, which is mounted and driven in tandem with the Maxaret unit, on the secondary casing of the gearbox, and draws its oil from the lubrication sump of this assembly. As a safety measure, the master cylinder is of the tandem design, one half being connected to the front brakes and the other to those at the rear. In the event of any failure in one system, the transmission control unit prevents locking of the wheels on which the braking is still effective, by transferring an appropriate proportion of the retarding effort to the other wheels. Thus, although the pedal load for a given rate of deceleration is increased, effective and safe braking on all four wheels is still available. If the servo pump should fail, the braking system remains positive, though the pedal effort is greatly increased. *Concluded*

Motorized Ventilators

TWO efficient motorized ventilator units for fitting to vehicles are marketed by Key-Leather Co. Ltd, 5 Urswick Road, London, E.9. Both units are of generally similar design but one is larger than the other: the larger version is intended for buses, coaches and the heavier types of commercial vehicles, and the other for caravans, minibuses and light commercial vehicles. In each case, the fan is driven directly by a 30 W motor, operating from a 12 V or 24 V supply; the larger unit has a six-blade fan and the smaller a five-blade one.

The motors are of the reversible type, so the ventilators can be used for either supplying or extracting the air. According to the manufacturers, the capacities of the two models are 250 and 90 ft^3/min. A series of mesh screens, embodied in the casing of each unit, prevents the entry of rain or foreign matter. The larger ventilator, which is available with either an adjustable or a fixed intake shutter, has an overall diameter of $15\frac{1}{4}$ in, and its depth is $5\frac{1}{8}$ in, of which only $\frac{7}{8}$ in projects into the interior of the vehicle; for the smaller unit, the corresponding dimensions are 8 in, 5 in and 1 in. A leaflet is available from Key-Leather Co. Ltd.

Stop Watch

FOR heavy-duty applications, a new 12 hour stop watch has been introduced by Ed. Heuer and Co. S.A, Biel-Bienne, Switzerland. The instrument is specifically designed for panel-board or dash mounting. It has an hour indicator disc, the numbers on which are viewed through an aperture, and the usual full-circle, centre-shaft, minute register. An alternative dial face, with extra peripheral divisions in hundredths of a minute, is available for process and research instrumentation, where decimal readings are required.

Start, stop and go-on control operations are effected by successive depressions of a knob on top. Return of the hands to zero is controlled by a push-button on the right and is operable only when the hands are at rest. On the left of the instrument is another push-button for zeroing the hour disc. This button is locked by a knurled nut to prevent its accidental operation. The design is such that the start, stop and go-on control can be operated also by a pedal or solenoid. The accuracy of the instrument is said to be within $\frac{1}{2}$ sec/hr. In the U.K. the sole distributors are Baume and Co. Ltd, 50 Hatton Garden, London, E.C.1.

Translations of German Standards

AMONG the English translations of German Standards, published in April by Deutscher Normenausschuss, Berlin W.15, Uhlandstrasse 175, are the following that are likely to be of interest to automobile engineers:

DIN UDC 621-229.2 Tool holders—138, TOOLS (MACHINE TYPE) FOR METAL WORKING: *bores keyways and driving features for tools with parallel bore and with 1 in 30 taper bore, September* 1955

UDC 621.83 Gears; toothed wheels—3971, GEAR TOOTH SYSTEMS: *specification factors and errors relating to bevel gears, basic terms and definitions, May* 1956

UDC 621.89 Lubrication; lubricants, oils and greases—51500, SHEET 2, LUBRICANTS: *procedure for determining demand and consumption, calculation procedure, January* 1959

Car Handling Characteristics

Directional Stability and Control of a Four-Wheel Vehicle in a Flat Turn

By MARTIN GOLAND and FREDERICK JINDRA*

The directional stability and control of a four-wheeled vehicle is studied by means of a simplified analysis which takes into account the variation of the cornering performance of the tyres with changes in their vertical loading. A detailed investigation is made of the steady, flat turn in order to appraise the effects of turn radius and forward speed on steering characteristics and dynamic directional stability.

Results of exploratory calculations are presented for a vehicle series in which the fore-and-aft location of the centre of gravity is varied; also, a variety of tyre configurations and inflation pressures are used on the front and rear wheels. It is shown that both turn radius and speed substantially affect the directional stability and control characteristics of the vehicle. The calculations also bring to light a variety of interesting conclusions regarding the influence of c.g. location, tyre tread width, and tyre inflation pressure on the directional performance of vehicles.

DURING a recent experimental study of the directional stability and control of automobiles executing steady flat turns, it was found that the observations could be closely correlated with the predictions from a relatively simple theoretical analysis. The experiments were concerned with the degradation of directional stability experienced, with certain automobile layouts, as a consequence of centrifugal acceleration in the turn, and the analysis affords a clear explanation of this phenomenon. The details of the theoretical study are presented here, with the suggestion that the results will be of value for preliminary design appraisals.

Turning performance of four-wheeled vehicles has been studied by several earlier authors. For example, the work of Paslay and Slibar[1], which is a comprehensive treatment of the problem, although not well adapted to the study of design trends, and that of Segel[2]. An extensive bibliography relating to directional stability and control problems for automobiles is contained in a paper by Milliken[3]. Also, a comprehensive bibliography on wheel shimmy, which contains material closely related to the present problem, has been prepared by Dengler, Goland and Hermann[4].

In the treatment presented here, the problem of directional stability and control in a flat turn is formulated in as simple a fashion as possible, consistent with an accurate description of the essential turning mechanisms. Thus, the dynamical equations of motion are written in two degrees of freedom, namely, vehicle yaw and vehicle sideslip. Rolling motions of the sprung mass are included as a quasi co-ordinate only, in order to enable calculation of the vertical forces acting at the tyres.

The conventional linear method of stability analysis has been adopted. First, the vehicle is visualized as travelling in an ideal, steady turn, and it is then disturbed slightly from this path. The degree of stability of such small, disturbed motions is then studied by means of small-disturbance, first order, theory.

It is evident that the cornering performance of the tyres plays a major role in the analysis. In order to trace the variations of cornering power for various tyres and for the different vertical load conditions encountered in turns, a series of semi-empirical equations, deduced by Smiley and Horne[5], are employed, but with some modification. The changes introduced in these equations are for the purpose of simplifying their forms—the consequent slight reductions in accuracy, as measured against the correlations of experimental data contained in the work by Smiley and Horne, do not appear to be objectionable for the purposes of this analysis.

*Southwest Research Institute, 8500 Culebra Road, San Antonio 6, Texas.

Equations of motion

It has already been mentioned that the dynamical degrees of freedom chosen for the analysis comprise vehicle yaw and vehicle sideslip. The steering system is eliminated from consideration as a degree of freedom, by assuming that the steering wheel is held fixed at a particular setting and that the steering linkage is rigid. Vehicle roll is not included as a dynamical co-ordinate, but will be introduced later as a quasi-static co-ordinate. Various other dynamic effects which are present in an actual vehicle are ignored in the interests of simplicity.

Let (x, y) be a set of cartesian axes fixed on the vehicle, and with origin at the vehicle c.g. The x-axis is longitudinal and the y-axis lateral; both are directed along principal axes of inertia of the vehicle—changes in vehicle configurations due to roll, etc, are neglected. From Fig. 1, the positive sense of the axes can be seen. Let $M = W/g$ be the total mass of the vehicle, where W is its weight and g is the acceleration due to gravity; the wheelbase length is l; the distance from the front axle rearward to the c.g. is a; and the lateral, track dimension between wheel centres is d. Further, let l_i be the radius of gyration that describes the vehicle polar moment of inertia about a vertical axis passing through the c.g.

Throughout the analysis, it is assumed that the c.g. of the vehicle is moving with the constant velocity V, as shown in Fig. 1. If the sideslip angle of the vehicle is β, with positive convention as shown in Fig. 1, then for small values of β the components of velocity along the x-axis is $V \cos \beta \simeq V$; in the y-axis direction the sideslip velocity is $V \sin \beta \simeq V\beta$.

The directional stability equations of motion for the vehicle can now be written in terms of the moving (x, y) axes. With the instantaneous yawing velocity notated ω, positive as shown in Fig. 1, the equations for a flat turn become

$$MV(\dot{\beta}+\omega)=F_1+F_2+F_3+F_4 \quad \ldots\ldots 1$$

$$M\ell_t^2\dot{\omega}=a(F_1+F_2)-(\ell-a)(F_3+F_4) \quad \ldots\ldots 2$$

where F_1, F_2, F_3 and F_4 are the lateral forces at the four wheels, positive in the same sense as the y-axis; the subscripts 1, . . . , 4 refer to the wheels as notated in Fig. 1; and a dot over a symbol denotes a time-derivative.

It is now pertinent to examine the nature of the cornering forces F. However, before this is done it is necessary to calculate the vertical loadings at the four wheels.

Vertical wheel-loadings

To maintain the vehicle in an ideal, steady, flat turn, the sprung mass must roll by a certain amount toward the outside of the curve; the extent of this roll will now be calculated. If slight disturbed motions are superimposed on the steady turn, the disturbances will, in general, consist of dynamically coupled vehicle yaw and sideslip, plus rolling and pitching motions of the sprung mass.

In the present analysis, the disturbed rolling and pitching motions of the sprung mass are not taken into account when calculating directional stability. This is permissible on the grounds that these motions exert only a minor influence on the directional stability characteristics of the vehicle, an assumption which appears justifiable.

Now suppose that the front and rear wheels are sprung elastically, and that k_f is the spring constant at each front wheel, with k_r the spring constant at each of the rear wheels. For an ideal steady turn with centrifugal acceleration V^2/R, a straightforward analysis of the equilibrium requirements leads to the conclusion that the outward body roll can be estimated from

$$\alpha=2\frac{W}{d(k_f+k_r)}\frac{h}{d}\frac{V^2}{gR} \quad \ldots\ldots 3$$

where h is the vertical distance of the vehicle c.g. above the ground.

The vertical force at each wheel is made up of the statical component, which acts when the vehicle is at rest, plus the component arising as a consequence of the body roll. Then, with $a'=a/\ell$

$$P_1=\frac{W}{2}\left\{(1-a')+2\frac{k_f}{k_f+k_r}\frac{h}{d}\frac{V^2}{gR}\right\} \quad \ldots\ldots 4$$

$$P_2=\frac{W}{2}\left\{(1-a')-2\frac{k_f}{k_f+k_r}\frac{h}{d}\frac{V^2}{gR}\right\} \quad \ldots\ldots 5$$

$$P_3=\frac{W}{2}\left\{a'+2\frac{k_r}{k_f+k_r}\frac{h}{d}\frac{V^2}{gR}\right\} \quad \ldots\ldots 6$$

$$P_4=\frac{W}{2}\left\{a'-2\frac{k_r}{k_f+k_r}\frac{h}{d}\frac{V^2}{gR}\right\} \quad \ldots\ldots 7$$

The unloading effects of the turn on the inside wheels is at once evident.

Cornering performance of the tyres

It is of interest to examine now the nature of the cornering forces F in the equations of motion, 1 and 2. Those who are familiar with the mechanics of pneumatic tyres will observe that equation 2 is not exact, since it does not take into account the aligning torque, or pneumatic caster, which acts at each wheel. However, for the purposes of this study, these torques are not significant.

The stability and control calculations here are concerned only with the circumstance when all four wheels have essentially rolling contact with the roadway. The onset of gross skidding at any one of the wheels represents a limiting condition beyond which the present analysis loses validity. However, more will be said about this in a later section.

It was remarked in the earlier discussion that the work of Smiley and Horne[5] forms the basis for the cornering force evaluations used in this study. The partly empirical equations they develop for the various tyre characteristics are not used directly. Rather, are their equations first simplified in a fashion that affects accuracy only slightly. For the sake of brevity, when the Smiley and Horne work is mentioned further, it will be assumed that a copy is available to the reader.

Consider first a rolling tyre without camber deflection, that is, the plane of the hub remains normal to the ground. Let ψ be the drift angle of the rolling tyre, that is, the angle between the plane of the hub and the direction of travel. Now define the parameter

$$\phi=\frac{N\psi}{\mu P} \quad \ldots\ldots 8$$

where P is the vertical load carried by the tyre, μ is the effective coefficient of friction between the tyre and ground, and N is the cornering power coefficient, to be defined later. Smiley and Horne—in their equation 79—then relate the cornering force F_ϕ, due to drift to the parameter ϕ, by the following equations

$$\left.\begin{aligned}\frac{F_\phi}{\mu P}&=\phi-\frac{4}{27}\phi^3\ ;\ \phi<1{\cdot}5\\ &=1\qquad\ ;\ \phi>1{\cdot}5\end{aligned}\right\} \quad \ldots\ldots 9$$

It is evident that complete skidding occurs when ϕ is greater than 1·5.

For the present analysis, the cubic term in equation 9 is disregarded. Then, with reasonable accuracy

$$\left.\begin{aligned}F_\phi&=N\psi\ ;\ \phi\leqslant 1\\ &=\mu P\ ;\ \phi\geqslant 1\end{aligned}\right\} \quad \ldots\ldots 10$$

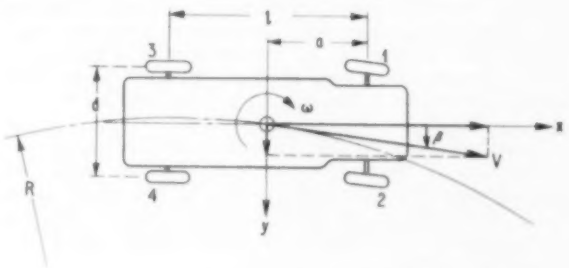


Fig. 1. Diagram showing the axial convention and symbols used for the calculations concerning the performance of a vehicle in a flat turn

In these simplified expressions, tyre skidding takes place when ϕ is greater than unity.

Smiley and Horne find that the value of the cornering power coefficient N is dependent on various tyre properties among which are the undeflected tyre diameter D, the maximum width w of the undeflected tyre, the rated inflation pressure p_r, the actual inflation pressure p, and the amount of the vertical deflection of the tyre hub Δ corresponding to the vertical load P. The vertical deflection Δ is also dependent on these same tyre characteristics.

An inspection of the data collected by Smiley and Horne shows that tyre bottoming generally occurs for values of Δ/w around 0·6. Until bottoming is approached, the relationship between P and Δ is roughly linear, save at small values of the ratio $\Delta:w$—see Fig. 13 and equation 23 in Smiley and Horne. For the present purpose, a linear relationship between P and Δ is presumed, and the bracket $(p+0{\cdot}08\ p_r)$ which appears in Smiley and Horne's equation 23 is replaced by $1{\cdot}08p$. Then, by fitting a curve to the data shown in their Figs. 13b and c, the following

approximate expression can be fairly readily obtained

$$\frac{\Delta}{D}=0{\cdot}42\left(\frac{D}{w}\right)^{\frac{1}{3}}\left(\frac{P}{pD^2}\right) \quad \ldots\ldots 11$$

The cornering power coefficient can now be written as a function of Δ/D. Smiley and Horne's equation 82 is used as the basis for this relationship, but simplified by setting $p \simeq p_r$. Then

$$\left.\begin{aligned}\frac{N}{pw^2}&=C_c\left\{1{\cdot}7\left(\frac{\Delta}{D}\right)-12{\cdot}7\left(\frac{\Delta}{D}\right)^2\right\};\ \frac{\Delta}{D}\leqslant 0{\cdot}088\\&=C_c\left\{0{\cdot}095-0{\cdot}49\left(\frac{\Delta}{D}\right)\right\};\ \frac{\Delta}{D}\geqslant 0{\cdot}088\end{aligned}\right\} \quad \ldots\ldots 12$$

where C_c is a coefficient, with a value of about 60 for modern types of tyres, and where the units for N are pounds per radian. A plot of equation 12, for $C_c=60$, is shown in Fig. 2 on this page.

The description of tyre mechanics thus far is based on the assumption that the plane of the hub remains perpendicular to the ground at all times, that is, the wheel has no camber deflection. If the wheel is in rolling contact with the ground and also has a camber angle of amount γ, then an additional side-force is induced. Smiley and Horne point out that only meagre data are available to describe the magnitude of this lateral force, and they suggest the very approximate relation

$$F_\gamma=-P_\gamma\gamma \quad \ldots\ldots 13$$

where the camber angle γ is taken as being positive in the same sense as the roll angle $\varkappa$, that is, the wheel tilts toward the outside of the curve. The constant P_γ has a value approximately equal to the vertical load P carried by the wheel.

Sufficient information is now available to enable the calculation of the lateral forces F at the wheels of the vehicle. For each wheel, the vertical load is first evaluated by means of equations 4-7. the ratio $\Delta:D$ is next computed from equation 11, and the value of N is determined from equation 12. From equations 10 and 13, the total lateral force F for rolling contact is then given by

$$F=N\psi-P_\gamma\gamma \quad \ldots\ldots 14$$

The calculation of the drift angle ψ is discussed later.

Equation 14 becomes invalid when the tyre skids. In the absence of camber, this occurs approximately when $\varphi=1$. With camber, the condition for the onset of skidding is modified in some fashion for which reliable data is not currently available.

It is to be noted that the relaxation length for the tyre is ignored in this discussion: while the effects of this phenomenon are important during rapid drift angle variations, such as occur during wheel shimmy, the stability motions of interest here involve relatively low frequencies of oscillation, and relaxation length effects are, of course, unimportant.

Tyre drift angles

Equation 10 indicates that, within the realm of rolling contact, the cornering force developed by each tyre is directly proportional to its drift angle ψ. The instantaneous drift angles at the four tyres are readily evaluated by kinematic considerations.

Consider, for example, the right front tyre, notated by subscript 2. A schematic sketch of the kinematics of this wheel is shown in Fig. 3. The forward velocity in the x-direction at the centre of the hub is equal to $V\cos\beta-(d/2)\omega$, as can be seen from Fig. 1. It has already been assumed for the purposes of this analysis that β is a small angle, and it will also be assumed that the ratio $\omega d/V$ is small. On this basis, the forward velocity in the x-direction at each wheel is, to a sufficiently close approximation, simply V.

The sideslip velocity at the right front tyre is seen from Fig. 1 to be $\beta V+a\omega$. The sideslip angle β_2 at the wheel is

$$\beta_2=\beta+\frac{a\omega}{V} \quad \ldots\ldots 15$$

Now, let δ_2 be the steering angle of the tyre. This steering angle may result from a steering wheel movement, a roll-steer suspension linkage, or a combination of the two. The positive sense for the angle δ is shown in Fig. 3. From Fig. 3 it can be seen that the drift angle for the tyre, that is, the angle between the hub plane and the direction of travel, is then equal to

$$\psi_2=\delta_2-\beta_2 \quad \ldots\ldots 16$$

or

$$\psi_2=\delta_2-\beta-\frac{a\omega}{V} \quad \ldots\ldots 17$$

It is to be noted that the cornering force equations given earlier, strictly speaking, refer to the lateral force normal to the wheel hub. Hence, the force F_2 acts in the direction shown in Fig. 3. However, if the steering angle δ_2 is small, so that $\cos\delta_2\simeq 1$, then the cornering force in the y-direction can be taken equal to F_2.

The drift angles at the remaining three wheels can now be deduced in similar fashion. They are readily shown to be as follows

$$\psi_1=\delta_1-\beta-\frac{a\omega}{V} \quad \ldots\ldots 18$$

$$\psi_3=\delta_3-\beta+\frac{(l-a)\omega}{V} \quad \ldots\ldots 19$$

$$\psi_4=\delta_4-\beta+\frac{(l-a)\omega}{V} \quad \ldots\ldots 20$$

where δ_1, δ_3 and δ_4 are the steering angles at the respective wheels. It should be noted that if wheel steering results from a roll-steer linkage, then the values of δ will be a function of the roll angle $\varkappa$, as determined by calculation from equation 3.

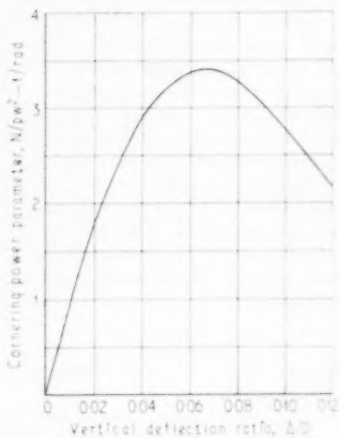


Fig. 2. Plot of cornering power parameter against the vertical deflection ratio, $C_c=60$

The ideal, steady flat turn

Before considering the directional stability of the vehicle, it is useful to calculate the steering wheel settings required to hold an ideal steady flat turn. In this calculation the question of vehicle stability is ignored and the vehicle is assumed to be capable of navigating continuously an ideal, circular path.

For a path of radius R and a vehicle speed of V, the vehicle yaw velocity is

$$\omega_0=\frac{V}{R} \quad \ldots\ldots 21$$

Also, for the ideal form of turn, it is clear that $\dot{\omega}=\dot{\beta}=0$.

For the calculation, the equations of motion, 1 and 2, must be combined with equations 14 and 17-21. If this is done, then after considerable algebraic manipulation, it is found that

$$\frac{\ell}{R}(1+KV^2)=\frac{\chi_{12}}{N_F}-\frac{\chi_{34}}{N_R} \quad \ldots\ldots 22$$

where

$$\chi_{12}=N_1\delta_1+N_2\delta_2-P_{\gamma 1}\gamma_1-P_{\gamma 2}\gamma_2 \quad \ldots\ldots 23$$

$$\chi_{34}=N_3\delta_3+N_4\delta_4-P_{\gamma 3}\gamma_3-P_{\gamma 4}\gamma_4 \quad \ldots\ldots 24$$

and where

$$N_F=N_1+N_2; \quad N_R=N_3+N_4 \quad \ldots\ldots 25$$

The terms $P_{\gamma 1}\gamma_1$ etc., refer to the lateral tyre forces due to roll-camber effects at the various wheels. The parameter K in equation 22 has particular significance and is defined as

$$K=\frac{M}{\ell}\left[\frac{(1-a')}{N_F}-\frac{a'}{N_R}\right] \quad \ldots\ldots 26$$

with $a'=a/\ell$.

To observe the role played by the parameter K, it is of interest to reduce equation 22 to the case of a vehicle without roll-steer and roll-camber suspension characteristics. Then

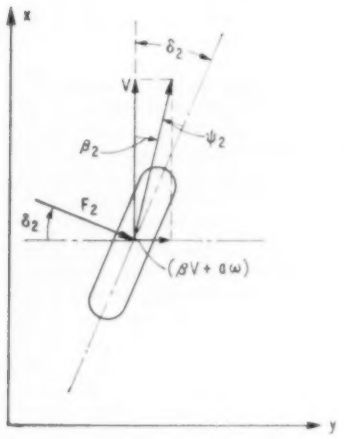


Fig. 3. Diagram illustrating the method of calculation of the drift angle of a right-hand front wheel of a car

$\delta_1=\delta_2=\delta$; $\delta_3=\delta_4=0$; and the various γ are zero. Then equation 22 becomes

$$\frac{\ell}{R\delta}=\frac{1}{1+KV^2} \quad \ldots\ldots 27$$

This is a well-known result, and shows that the vehicle will understeer whenever $(1+KV^2)>1$; it will oversteer if $(1+KV^2)<1$. The control reversal point is determined by the condition $(1+KV^2)=0$.

It is of particular importance to note that, even for a given vehicle, the parameter K cannot be thought of as remaining constant. It will vary with the ratio V^2/gR, which in turn fixes the vertical loads at the four wheels and hence the values of the cornering coefficients. Sample calculations will be presented later to show that the variations in K are truly significant within the operating ranges of many vehicles.

For reasons of brevity, the analytical expression for the sideslip angle β is not given here. However, it can be derived in the same manner as was that in equation 22.

Directional stability in the turn

The preceding analysis of the steady turn does not take into account the stability of transient motions of the vehicle, such as those due to road imperfections, wind gusts, and so forth. An unstable vehicle, or one which is marginally stable, will require considerable skill on the part of the driver to maintain control.

For the study of directional stability, it is first imagined that the vehicle moves in an ideal, steady turn. The vehicle is then given a small disturbance and the resultant transient motions are studied. If the transient motions are damped and die away, the vehicle is stable; if they diverge and grow larger with time, the vehicle is unstable.

The method of analysing for the small-disturbance, linear stability of systems such as equations 1 and 2 is well-known and involves arriving at the complementary solution to the equations of motion. Equations 14 and 17-20 are first inserted in equations 1 and 2. It should be noted that, within the assumptions of the present study, variations in the roll angle α, and hence the camber angles γ, during the small-disturbance motions are neglected. The equations of motion that express the nature of the transient motions are then

$$MV\dot{\beta}+A_{11}\beta+A_{12}\Omega=0 \quad \ldots\ldots 28$$

$$A_{21}\beta+MV\left(\frac{\ell_t}{\ell}\right)^2\dot{\Omega}+A_{22}\Omega=0 \quad \ldots\ldots 29$$

where

$$\Omega=\frac{\omega\ell}{V}$$

and where

$$A_{11}=N_F+N_R \quad \ldots\ldots 30$$

$$A_{12}=\frac{MV^2}{\ell}+a'N_F-(1-a')N_R \quad \ldots\ldots 31$$

$$A_{21}=a'N_F-(1-a')N_R \quad \ldots\ldots 32$$

$$A_{22}=(a')^2N_F+(1-a')^2N_R \quad \ldots\ldots 33$$

Transient solutions for the unknowns β and Ω are now sought in the functional form $e^{\lambda t}$, where t is the time variable. The characteristic equation for λ is then found, from equations 28-33, to be

$$(V\lambda)^2+\eta(V\lambda)+\zeta^2(1+KV^2)=0 \quad \ldots\ldots 34$$

where K is given by equation 26, and where

$$\eta=\frac{1}{M}\left\{(N_F+N_R)+\left(\frac{\ell}{\ell_t}\right)^2[(a')^2N_F+(1-a')^2N_R]\right\} \quad \ldots\ldots 35$$

$$\zeta^2=\left(\frac{\ell}{\ell_t}\right)^2\frac{N_FN_R}{M^2} \quad \ldots\ldots 36$$

The two values of λ corresponding to equation 34 are then given by

$$2V\lambda=-\eta\pm[\eta^2-4\zeta^2(1+KV^2)]^{1/2} \quad \ldots\ldots 37$$

and it is these λ values which determine the directional stability of the vehicle in the circumstances considered.

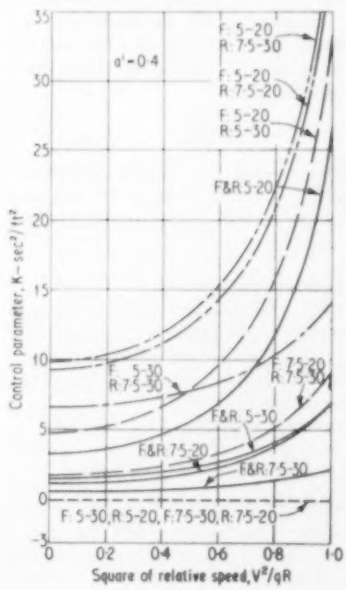
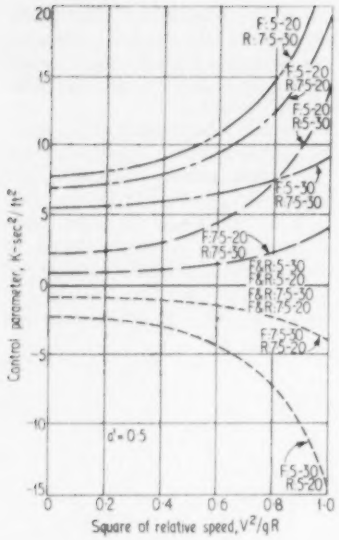


Fig. 4. Curves showing the variation of the control parameter K plotted against V^2/gR, where $a'=0{\cdot}4$



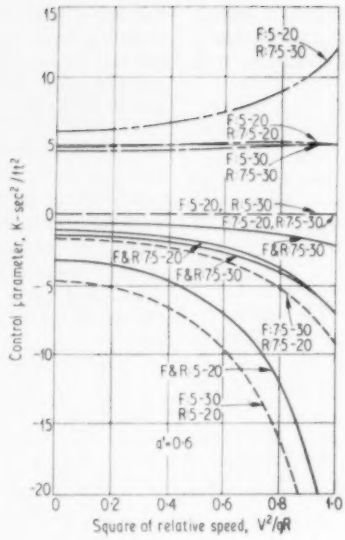


Fig. 5, far left: Variation of control parameter K plotted against V^2/gR, $a'=0{\cdot}5$

Fig. 6, left: Variation of the control parameter K plotted against V^2/gR, $a'=0{\cdot}6$

Several significant conclusions can at once be drawn from equation 37. To begin with, it is to be noted from equations 35 and 36 that both η and ζ^2 always have positive values. Hence, according to the value of $(1+KV^2)$, the following conclusions are applicable:

(a) For $(1+KV^2)>\eta^2/4\zeta^2$ the square-root bracket in equation 37 has an imaginary value. Hence, the system has *oscillatory stable* directional stability.

(b) For $0<(1+KV^2)<\eta^2/4\zeta^2$ the square-root bracket is positive, but with a value less than η. Hence, both values of λ are real and negative, and the system has *dead-beat stability*.

(c) For $(1+KV^2)<0$, the square-root bracket is positive and has a value greater than η. Hence, one value of λ is real and positive, and the vehicle is directionally *unstable*.

The condition $(1+KV^2)=0$ represents the condition for neutral directional stability, and hence separates the regions of stable and unstable operation. It will be recalled from the earlier discussion that this same condition defines the control reversal point for the vehicle.

Because of the significance of the condition $(1+KV^2)=0$ as an index of both control and directional stability characteristics, it is useful to define the *critical velocity* for the vehicle as

$$V_{cr}=\left(-\frac{1}{K}\right)^{\frac{1}{2}} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 38$$

It is evident that V_{cr} has real values for only negative values of K. Equally clear is the fact that the value of V_{cr} should be kept well above the operating range of the ordinary vehicle.

Sample calculations

To illustrate the nature of the results that follow from the analysis, numerical calculations have been carried out for a hypothetical series of vehicles. For the entire series, certain numerical parameters are held constant. These are as follow:

$W=2{,}500$ lb; $\quad l=9$ ft;

$\frac{h}{d}=0{\cdot}5$; $\quad l_1=\frac{l}{2}$

To reduce further the number of variables in the calculations, it is assumed that the vehicles have no roll-steer or roll-camber suspension characteristics.

One of the important parameters, which influences vehicle steering and directional stability, is the longitudinal placement of the vehicle c.g. Hence, three values for the parameter a' are used in the calculations, namely, $a'=0{\cdot}4$, $0{\cdot}5$ and $0{\cdot}6$. The simplest conception of a suspension system is that in which each wheel is sprung by means of a vertical, linear spring; of the design requirement is stipulated that the chassis must remain level for all values of the gross weight W, then certain relations must hold between the spring stiffnesses at the front and rear wheels. Employing this simple conception for vehicle design, it can then be readily shown that for

$$a'=0{\cdot}4, \quad \frac{k_f}{k_r}=\frac{3}{2}$$

$$a'=0{\cdot}5, \quad \frac{k_f}{k_r}=1$$

$$a'=0{\cdot}6, \quad \frac{k_f}{k_r}=\frac{2}{3}$$

and these values are employed in the present calculations.

Another factor of considerable design interest is the influence of tyre characteristics on directional stability and control. The calculations were, therefore, arranged to show the effects of various tread widths and tyre pressures at the front and rear wheels. In particular, the wheel diameter D was kept at 24 in for all wheels, but the tyre width was taken as either 5 in or 7·5 in, and the tyre pressure was set at either 20 lb/in² or 30 lb/in². A variety of different front and rear wheel configurations were used on each vehicle, and these are described by means of the following code in the results presented later. For the front wheel description, the letter F is followed by a number designating the tyre width and by a second number specifying the inflation pressure. For the rear wheels, the letter R is used in similar fashion. Thus, the designation F:5-30 refers to front wheels of 24 in diameter, with a 5 in tyre width, inflated to a tyre pressure of 30 lb/in².

The calculations are primarily aimed at showing how the vehicle stability and control characteristics change when steady flat turns are traversed at various speeds and curvatures. For the present study, the turn radius was kept constant at the value $R=150$ ft, which represents a reasonably tight turn. The forward speed of the vehicle was a variable parameter.

The manner in which the calculations are carried out is evident from the analysis presented earlier. For each

driving condition, the vertical wheel-loadings are first computed from equations 4-7. The tyre deflections are then calculated from equation 11, and the value of N for each tyre is determined from equation 12. Provided that skidding does not occur, the value of the control parameter K can be calculated from equation 26. The steering angle δ required to hold the vehicle in the turn then follows from equation 27. Finally, the dynamic stability of the vehicle can be appraised by calculating the stability roots λ from equation 37.

Shown in Fig. 4 are the values of K for the vehicle with $a'=0{\cdot}4$ and equipped with various configurations of tyres, and plotted as a function of the parameter V^2/gR. This particular abscissa is chosen because the value of K does not depend on the individual choices of V and R, but rather on the ratio of the centrifugal acceleration V^2/R to the gravity acceleration g.

A zero value of K corresponds to perfect geometric steering performance by the vehicle. It is of interest to note that this condition is achieved in Fig. 4 for all values of V^2/gR, provided the front and rear tyres are identical, but a 30 lb/in^2 inflation pressure is used at the front and a 20 lb/in^2 inflation is used at the rear wheels.

It follows at once from equation 27 that a positive value for K is indicative of an understeer control characteristic, whereas a negative K value results in an oversteer control characteristic. For the forward c.g. location, $a'=0{\cdot}4$, Fig. 4, it is seen that an understeer performance is obtained for all the tyre configurations chosen, except in the instances where perfect geometric steering is attained. It is of particular interest, however, that there is a strong dependence of K on the V^2/gR parameter.

Shown in Fig. 5 is the variation of K for $a'=0{\cdot}5$, that is, when the longitudinal position of the c.g. is midway between the front and rear axles. Obviously, when all four wheels are identical in tread and in inflation pressure, perfect geometric steering results; however, when the front tyre pressure is higher than the rear, an oversteer characteristic is, of course, encountered, and this becomes more pronounced as V^2/gR increases. For a rear inflation pressure higher than that at the front, an understeer control increasing with V^2/gR is encountered. The effect of tread width is also evident from the various cases plotted in Fig. 5. Once again, the strong variation in the values of K with V^2/gR is emphasized, as shown by the curves.

In Fig. 6, the rearward position of the c.g, $a'=0{\cdot}6$, is illustrated. It can be seen that special precautions must be taken if an understeer control characteristic is to be maintained. For all four tyres identical in tread and inflation pressure, oversteering will result. Perfect geometric steering can be achieved with identical tyre treads by inflating the rear tyres to a higher pressure than the front. Interestingly, understeer control can be accomplished by using rear tyres with a wider tread width than at the front.

The steering characteristics displayed in Figs. 4-6 are further clarified in Figs. 7-9. In these plots, a turn radius of 150 ft is assumed, and the abscissa represents forward speed. The ordinate employed is δ/δ_o, that is, the ratio of the steering angle required to hold the vehicle in the turn to the perfect geometric steering angle, that for $V=0$. From Fig. 7 it can be seen that, for $a'=0{\cdot}4$ increasing steering angles are, in general, required at the higher speeds. For $a'=0{\cdot}5$, Fig. 8, control reversal can occur within the driving speed range, when the rear tyres are under-inflated relative

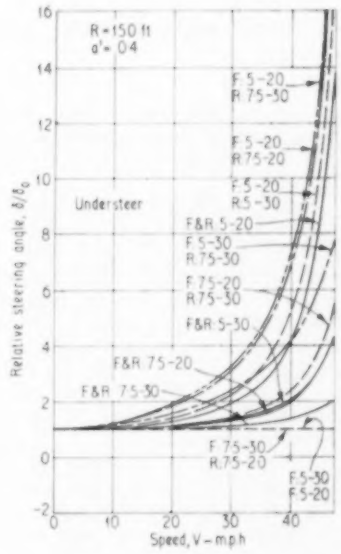
Fig. 7, right: Relative steering angle necessary to hold the vehicle in a steady flat turn—plotted against forward speed—where $R=150$ ft and the value $a'=0{\cdot}4$

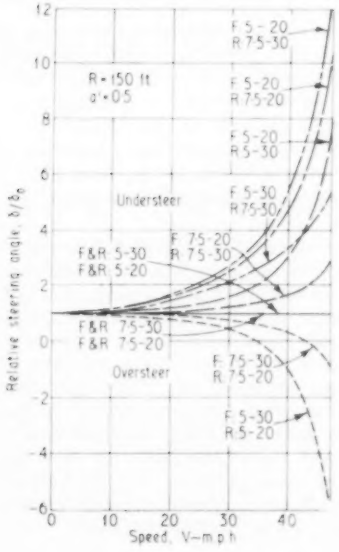
Fig. 8, below left: A plot, as in Fig. 7, but with the value of $a'=0{\cdot}5$

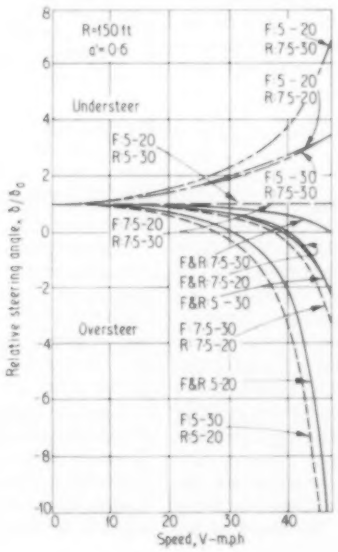
Fig. 9, centre: A plot, as in Fig. 7, but with the value of $a'=0{\cdot}6$

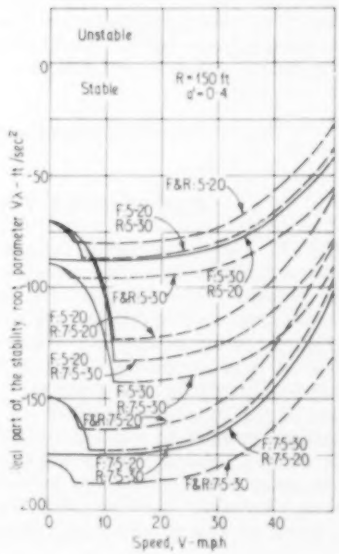
Fig. 10, below right: The real part of the stability root parameter plotted against the forward speed of a vehicle in a steady flat turn, where $R=150$ ft and the value $a'=0{\cdot}4$

——non-oscillatory mode,
— — — oscillatory mode









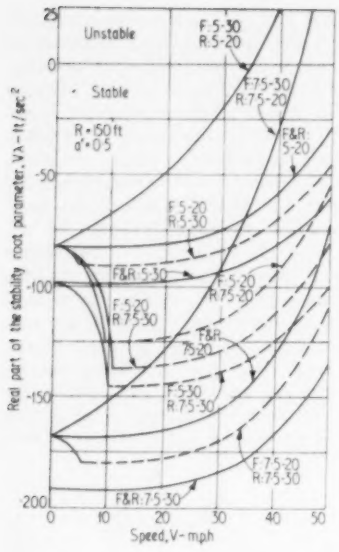
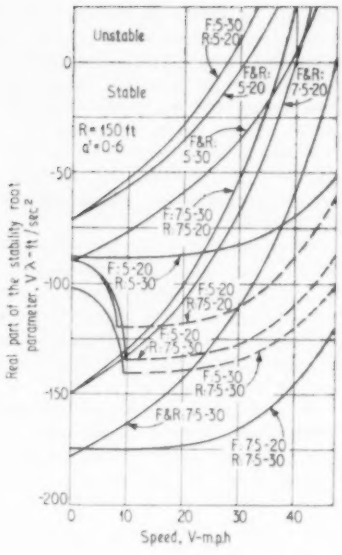


Fig. 11, left: These curves show the real part of the stability root parameter plotted against the forward speed of a vehicle executing a steady flat turn, where R=150 ft and the value a'=0·5

Fig. 12, right: As above, illustration of real part of the stability root parameter plotted against the forward speed of a vehicle executing a steady flat turn, where R=150 ft but the value a'=0·6

—— non-oscillatory mode,
— — — oscillatory mode



to those at the front. That there is a strong tendency toward control reversal with increasing speed for vehicles with a rearward location of the c.g, $a'=0{\cdot}6$, is shown in Fig. 9.

Finally, consider Figs. 10-12, which show plots of the *real part* of the stability root parameter $V\lambda$ as a function of the forward speed for the 150 ft turn. The solid lines in these plots indicate regions where the roots λ are real, and the values shown are for the least stable of the two non-oscillatory, dead-beat modes. Indicated by the dotted lines are regions where the roots λ are complex, denoting oscillatory stability. Only the real part of the parameter $V\lambda$ is plotted, but complete calculations show a steady decrease in the stability of the oscillatory mode as the speed is increased.

For $a'=0{\cdot}4$, Fig. 10 shows the vehicle to be dynamically stable over the entire speed range studied. With $a'=0{\cdot}5$, Fig. 11, dynamic instability within the driving speed range can occur when the front tyres are over-inflated compared to those at the rear. For $a'=0{\cdot}6$, Fig. 12, dynamic instability is introduced at surprisingly low values of forward speed when the rear tyres are under-inflated. Over-inflating the rear tyres relative to those at the front is seen to be a strong stabilizing factor for the vehicle with rearward location of the c.g. The very strong influence of both speed and turn radius on the stability performance of the vehicle is clearly demonstrated in Figs. 10-12. One of the principal purposes of the present paper is to emphasize and clarify the nature of this dependence.

The authors wish to thank Mr. John M. Clark, director, Automotive Products and Equipment Research, Southwest Research Institute, for his invaluable aid in the preparation of this paper. His design background and practical grasp of automotive performance features contributed greatly in the development of the ideas presented here.

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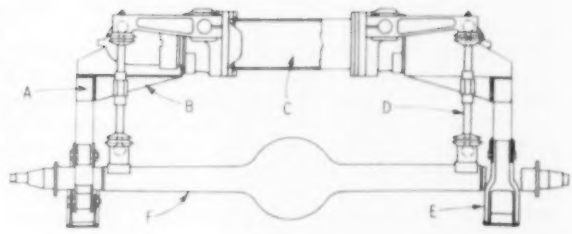
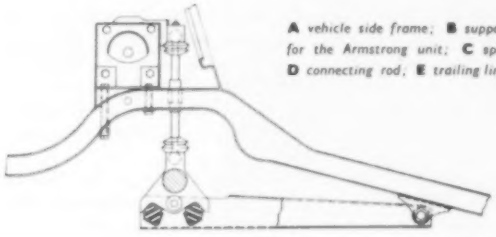
Apprentice Training

UNTIL now, many small companies, because they have lacked facilities, have been unable to accept apprentices. It is obvious, however, that by grouping together, and exchanging apprentices for appropriate periods, firms of this type would be able to provide a training comparable to that of large companies. In fact, a number of advantages can be claimed for such a scheme. When an apprentice in one of the larger firms finishes his time, he naturally feels inclined to move on and test his skill against new challenges in other companies. With a group scheme, however, the youngster works out his wanderlust while he is serving his time. Moreover, in moving from firm to firm, the boy gains a social as well as a technical education. He also has the opportunity of experiencing the different conditions in the firms in which he works.

An apprenticeship scheme of this type, especially designed to assist small companies, is run by a firm of management consultants, Industrial Administration (R and A) Ltd, 18 Thurloe Place, London, S.W.7. The scheme is called the Engineering Industries Association Group Apprenticeship, which is abbreviated to EIGA. There are now 22 groups, in the London, Birmingham, Manchester and Gloucester areas. In each group there are between 20 and 30 firms and up to 60 boys. In total, the scheme now provides 1,000 places, and about 380 firms are involved. Detailed syllabuses have been prepared for 42 different trades, to ensure that each boy is given a sound training, attends the most suitable courses and examinations.

The illustrations to the right and at the foot of this page are sketches showing various installation arrangements of the Armstrong suspension unit. Those on the right are two views of a rear axle layout; the trailing links are flexibly attached to the axle to give torsional resilience, for asymmetric movements; the panhard rod is not shown. Of the sketches below, which of course are not intended to be accurately representative of a suspension geometry, the most compact is that on the left. For a rear axle, a relatively light leaf spring could be used for location and to take the static unladen weight, while the Armstrong suspension unit would compensate for the variations of the remainder of the load





A *vehicle side frame;* **B** *support bracket for the Armstrong unit;* **C** *spacer tube;* **D** *connecting rod;* **E** *trailing link;* **F** *axle*

The Armstrong Suspension Unit

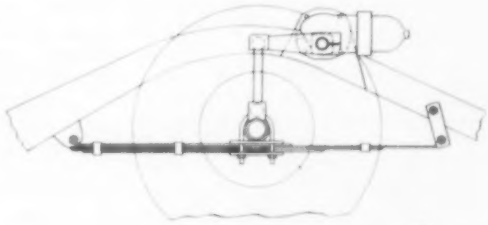
Compact and Completely Self-Contained Damper and Air Spring Assembly for Cars

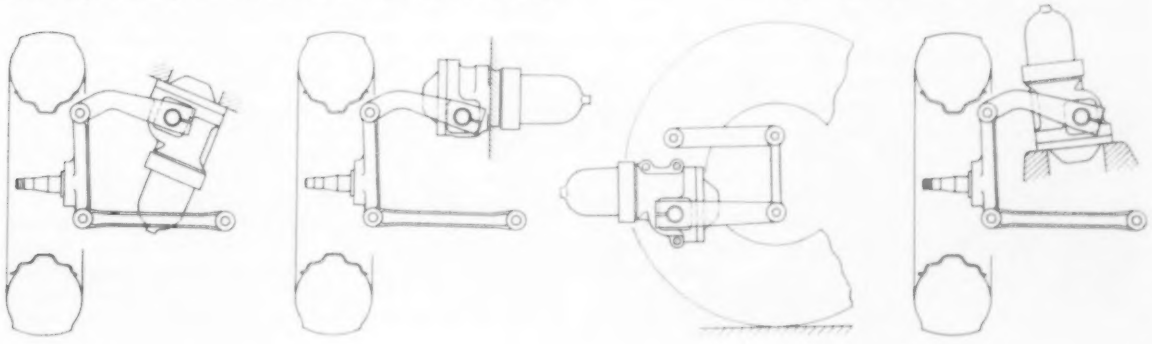
DESPITE its considerable advantages, air suspension has not yet made much progress, especially for cars and light commercial vehicles. The main reasons for this are, of course, the complexity of most of the air suspension systems that have been offered so far and also the servicing problems that arise, particularly with regard to the levelling valves and the freezing of condensation in them during cold weather. Clearly, however, in view of the advantages of air suspension, there has been a strong incentive for the development of a system in which these shortcomings are obviated.

The principal advantages inherent in all of the systems are as follows. By virtue of the employment of air as the springing medium, the body of the vehicle is effectively isolated from irregularities of the road surface. Nor is it subjected to secondary disturbances arising from either uncontrolled friction characteristics, as in the case of a leaf spring, or high frequency vibrations, which are likely to be experienced in torsion bar and coil springs. In some instances, too, benefits can be obtained by virtue of the inherent variable rate characteristics of air springs; this potential advantage cannot always be fully realized, however, in practice. Perhaps most important of all is the fact that the height of the sprung mass relative to the axles can be readily adjusted automatically, to compensate for variations in the load carried by the vehicles.

It would appear that the Armstrong hydro-pneumatic suspension unit has all of the advantages of air suspension, with none of the disadvantages mentioned above, nor is there any tendency for the vehicle to settle down on to the bump stops when it is standing, for example, overnight. Despite the fact that the unit currently available is in prototype form, for development work, and is therefore generously proportioned, it is strikingly compact and self-contained. Nitrogen gas is used as the springing medium, and it is in a sealed compartment. It follows that external components, such as compressors, pipelines and accumulators, have been obviated.

Levelling is effected hydraulically. Again, since the necessary pressure in the hydraulic fluid is built up by means of a pump incorporated within the unit and actuated by the suspension motion, there are no external components. All the levelling operations are effected automatically. From the accompanying illustrations it can be seen that one of the biggest advantages offered is the facility with which the unit can be assembled into an otherwise conventional suspension system, provided the vehicle structure has been designed for, or can be strengthened locally for, the mounting of the unit. The actual attitude in which it is mounted is





immaterial, because the hydraulic section of the unit is completely filled with oil and all air excluded.

Among the incidental advantages are the fact that the whole mechanism is enclosed in a strong casing, so it is completely protected from dirt and damage by, for example, stones flung up from the road. All the moving parts of the mechanism are completely immersed in oil and are, therefore, well lubricated. This, and the fact that the damper piston is unusually big, and therefore relatively lightly stressed in terms of load/in^2 of bearing area, should ensure long life.

In certain circumstances, it might be desirable to use the Armstrong suspension unit in conjunction with, for instance, a leaf spring: normally, the steel spring would be designed to take the static weight of the unladen vehicle, while the Armstrong unit would afford the additional support needed both to carry the extra loads and to effect the automatic levelling.

Development of the unit has been going on for four or five years. At the outset, the possibility of adapting telescopic type dampers for the purpose was considered carefully, but a spring-damper assembly in this form has a number of disadvantages: for example, the seal has to operate under conditions of sliding instead of rotary motion. This entails problems not only with regard to keeping oil in but also keeping foreign matter out. A telescopic unit tends to be large and therefore difficult to accommodate; moreover, it adds appreciably to the unsprung weight. Additionally, the arrangement of the levelling valves is more difficult.

Eventually, therefore, the lever type of unit was chosen. It represented a relatively easy step from the hydraulic suspension-levelling device, described in the April 1961 issue of *Automobile Engineer*. With the leverage practicable with this type of unit, it is not difficult to obtain high pressures, and compactness is therefore easy to achieve. Everything can be self-contained in a single casing, and all moving parts can be completely immersed in hydraulic fluid. Because of the favourable sealing conditions, and therefore the unlikelihood of the entry of grit, the finish of the cylinder bore can be retained almost indefinitely. Finally, the lever arm can serve the dual function of operating the unit and helping to locate the suspension assembly.

General description

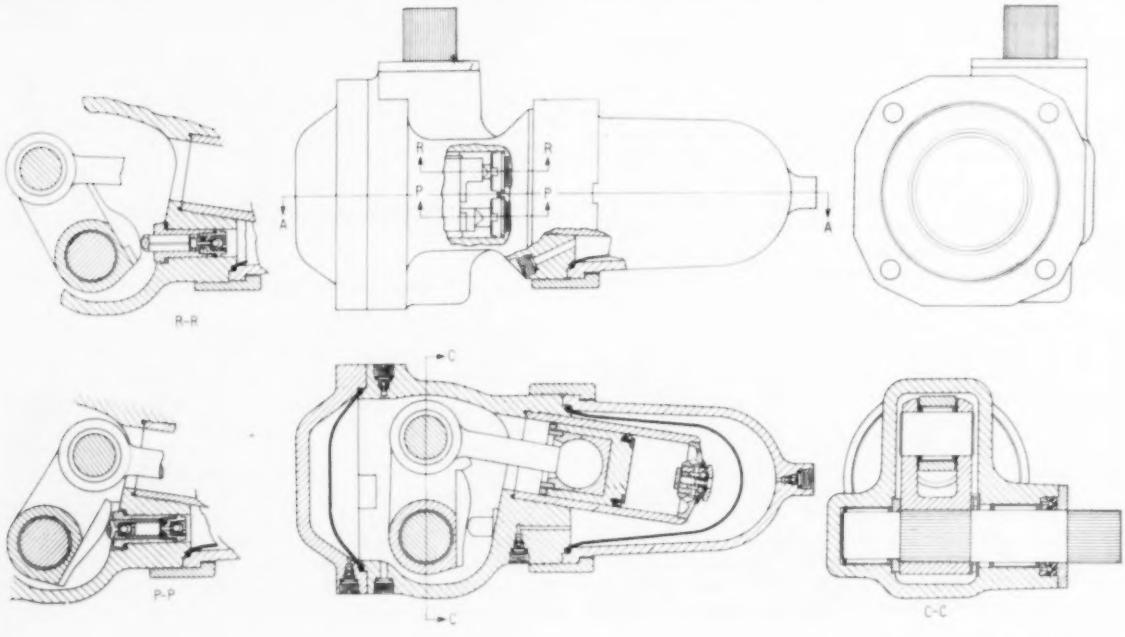
Essentially, the basic mechanism for the operation of the unit is a crank, or rocker, one arm of which is connected to the suspension and the other arm to the damper piston. As can be seen from the accompanying illustration, the cross-section is not unlike that of a single cylinder internal combustion engine. The crank housing and the cylinder of the damper assembly, together with two flexible diaphragms, one at each end, form an enclosure that is completely filled with hydraulic oil. One of the diaphragms seals the open end of the crank housing, while the other, which is deeply cupped, fits over the whole of the damper cylinder assembly. On the outer side of each diaphragm is nitrogen gas, under pressure, in a chamber formed by an appropriately domed, metal end-cap.

The end cap over the crank housing can be likened to the sump, and the nitrogen gas that it contains serves the purpose of giving the spring a rapidly falling rate as the full rebound position is approached. The domed cap over the damper piston and cylinder assembly at the other end is, of course, much larger and forms the main chamber for the springing medium.

Damper valves are incorporated in the cylinder head, which is screwed into the end of the cylinder. The pump and release valves for automatic levelling are housed in a position corresponding to that of the tappets in an internal combustion engine: that is, in the crank housing casting, adjacent to the end of the skirt of the cylinder. They are actuated by cams on the crank arm that is connected to the piston. At this point, the resemblance to the internal combustion engine ceases, because there are, of course, no push rods or other connection between them and the cylinder head.

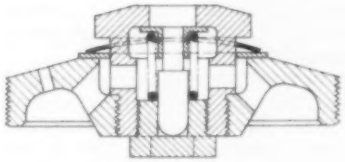
Before describing the design in greater detail, it must be added that, since this is a protoype unit designed solely to prove and develop the basic essentials of this new system,

By setting the damper cylinder at an angle relative to the longitudinal axis of the main casing assembly, optimum torque capacity has been obtained



the mechanical components have been made intentionally of unnecessarily robust form. The main reason has been that, at the outset, the unknown factors were those concerning the hydraulics and, to a lesser extent, the pneumatic springs. It was, therefore, felt to be desirable for the mechanical elements to be designed to give the utmost rigidity, reliability, and freedom from wear regardless of the length and severity of the development tests to be undertaken. Otherwise, the work might have been interrupted, perhaps at inconvenient junctures, by the need for replacing components.

Right: Noteworthy features of the Armstrong combined spring and damper unit are its compactness and the fact that it is completely self-contained



Left: Details of the damper valve and orifice arrangement in the head of the cylinder

Details of design—crank assembly

Crank arms of En.8 are splined on to an En.36 rocker shaft. This shaft is rated for a nominal load-carrying capacity of 9,000 lb-in torque, and its bearing portions are case hardened. On each side of the splined portion that carries the crank is a Torrington needle roller journal bearing in a boss in the crank housing, the bearing at one end being in a blind hole in the boss, while that at the other end is in a through-hole, beyond which projects the end of the shaft that carries the external, actuating arm. At the outer end of this hole is a counterbore to receive a synthetic rubber, chevron type seal, with a plastics spreader ring. The seal assembly is retained by an end-plate, spigoted into the counterbore and secured by four set-screws to the outer face of the boss. This seal should give no trouble since it is subjected to only the low pressure in the crank housing.

Since one of the bosses has a blind hole and the other a through-hole, and the shaft is subjected to hydraulic pressure, there is an axial thrust to be accommodated. Therefore, a Torrington needle roller thrust bearing is interposed between the crank web and the boss containing the through-hole. A phosphor bronze spacer ring is interposed between the other face of the crank web and the end of the blind boss. The crank web is an interference fit on the splines, and is positively positioned by a grub screw in two holes aligned with one another in the crank boss and shaft.

The two cams for actuating the levelling valves are of En.11, flame hardened to Rockwell C 60. An alternative that is under consideration is En.33, case hardened. These cams are secured to the crank by means of socket screws. Their bearing areas are large and the loading is light, so no trouble is experienced with regard to wear.

At the other end of the crank arm, which is forked, is a $1\frac{1}{8}$ in diameter, case hardened En.36 pin that carries the big-end of the En.8 connecting rod that actuates the damper piston. Torrington needle roller bearings, 1 in long, are employed in the big-end, because it is relatively heavily loaded. The connecting rod is about $\frac{3}{4}\frac{1}{4}$ in diameter. Its small end is of spherical form, because a ball-and-socket type joint is employed instead of the gudgeon pin.

Damper piston-and-cylinder assembly

An Oilite cup component is used in the ball-and-socket joint of the small end. The part-spherical retaining collar is of phosphor bronze, and the whole assembly is secured by a ring nut screwed into the skirt of the cast iron piston. A lock nut is screwed on to the ring nut and tightened against the end of the piston skirt.

The crown of the piston is shouldered to carry the piston ring, or seal, which is retained by an end cap secured by countersunk set-screws to the crown. As can be seen from the accompanying illustration, the details of this seal, which is of synthetic rubber, are unusual. It is of U-form and its inner periphery and face that seats on the shoulder are bonded to a flanged steel collar. This collar is essential to afford adequate support to the seal. A groove is machined in the seating face of the shoulder, to carry an O-ring. When the retaining cap is tightened down on to the piston crown, it pulls the seal hard against this O-ring, thus preventing leakage of fluid past the flanged steel collar.

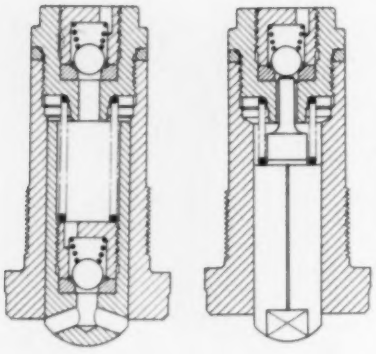
Obviously, a great deal of development work has had to be done on this sealing arrangement. The main problem has been to obtain an adequate seal with the minimum of friction. Apart from the details already mentioned, the actual shape and depth of the U-channel in the seal are critical. If the friction between the seal and the cylinder bore is too high, problems are likely to arise, owing to the generation of heat locally within the sealing material, and therefore wear; also, any tendency for the piston to stick will adversely affect the springing and damping action.

As can be seen from the illustration, the cylinder is set with its axis at an angle of about 8 deg to that of the domed steel cover in which it is enclosed. In an earlier design, the axes were parallel, but it was found that by canting the cylinder over in this way the torque capacity of the unit could be increased by about 25 per cent by virtue of the larger leverage obtainable.

The cylinder is a simple mild steel sleeve with a chromium plated bore. It is threaded near the end of its skirt, where it is screwed into the LM-16 aluminium alloy crank housing. A shoulder is machined on the lower end of the skirt to receive a synthetic rubber sealing ring for preventing leakage of fluid from the low pressure to the high pressure sides of the damper assembly. In production, the bore of the cylinder is first honed, then chromium plated and, finally, given a light honed finish.

At the other end of the cylinder is the screwed in mild steel cap, in which are incorporated the damper valves. As can be seen from the accompanying illustration, in this cap there is a small bleed hole, or holes, according to the degree of damping needed, through which oil can pass in either direction. This ensures that, under static conditions, the pressure is equalized on both sides of the cap, and the weight of the vehicle therefore supported by the action of the pressure in the gas, transmitted through the hydraulic fluid to the piston.

Screwed in the centre of the cap are the bump and rebound damping valves. These valves are coaxial. The



On the extreme left is the pumping valve, while the other illustration is of the release valve, with the restriction groove machined along the largest diameter portion of its plunger

central one, seated by a compression coil spring, opens during the rebound stroke, whereas the plate valve around the periphery of the assembly is seated by a disc type spring similar to a Belleville washer, and it opens during the bump stroke. Since the arrangement is clear in the illustration, a more detailed description is unnecessary. So far as the damping characteristics are concerned, normal practice is followed in that more damping is applied during the rebound than the bump stroke.

An oil resistant rubber to B.S.2751 is used for the cupped diaphragm over the cylinder assembly. Material to the same specification is employed for the diaphragm between the crank housing and the low pressure gas chamber at the opposite end of the unit. The periphery of each diaphragm is clamped between its domed end cover and the crank housing. Eight set-bolts secure the low pressure cover, while the cold-extruded steel high pressure one is retained by a ring nut. The main requirement for steels used in high pressure applications is, of course, ductility. If a brittle material is employed there is a danger that, in the event of an abnormally rapid rate of pressure build-up, fracture might occur. Since the pressure in the other chamber is only about 100 lb/in^2, LM-16 aluminium alloy is adequate.

Pumping and release valves

The automatic levelling function is performed by a pair of valve assemblies, actuated by the two cams on the crank arm. One of these assemblies is, in fact, a small pump containing two ball type non-return valves, while the other is simply a push rod actuating a single, ball type release valve.

Movements of the suspension system cause the plunger of the pumping valve to reciprocate. The arrangement of this valve is shown in the illustration. During the bump stroke, the plunger is moved upwards, the lower of the two ball valves closes and the hydraulic fluid above it is forced out past the other ball valve into the hydraulic chamber surrounding the damper cylinder and piston assembly. On the rebound stroke, the plunger is returned under the influence of the coil spring and, while the upper of the two ball valves is seated, the lower one opens to alow hydraulic fluid to enter from the crank housing to replace that which was forced out during the bump stroke.

Hitenspeed steel, supplied by British Rolling Mills Ltd, of Brymill Steel Works, Tipton, Staffs, is used for the barrel in which the plunger operates. This material can be machined easily and has good wear resistance. The bore is hone finished. Mild steel is used for all the screwed in components, and the valve seats are of nylon. A cyanide hardened En.202 plunger is employed.

For the release valve, the materials employed are similar to those of the pumping valve. It is a much simpler component, the construction of which can be seen in the illustration. When the push rod is lifted by its actuating cam, it unseats the ball valve, allowing fluid to pass from the high pressure chamber around the damper cylinder into the low pressure, crank housing. In doing so, the fluid passes through the valve and down a small groove machined the full length of the push rod. The function of this groove is to restrict the flow so that the levelling operation is not performed so rapidly as to upset the trim of the vehicle as it passes round a curve.

Filling

A measured quantity of SAE 5 hydraulic fluid, which has a pour-point of −40 deg F, is put into the space between the diaphragms during assembly. The system is bled and should require no further attention during service. The nitrogen gas is put into the gas chambers during assembly, and the filler valves sealed. It is not expected that further replenishments of the gas will be necessary during the life of the unit. The actual length of life cannot yet be definitely quoted. However, development experience indicates that there is every reason to believe that the life should be indefinite. Operational experience will, of course, be necessary to prove this conclusively.

It is hardly likely that a garage would have the necessary equipment for replenishment with either hydraulic fluid or gas, should this be necessary after a very great mileage. However, filling is a relatively simple operation for the manufacturers or a specialist service depot. All that is needed is a jig, a high pressure cylinder of nitrogen gas—readily obtainable commercially—a special container, of known volume and fitted with a pressure gauge, pipelines and cocks. The cylinder of gas is connected to the container, which in turn is connected to the chamber of the suspension unit. A cock is fitted in each pipeline.

Since the volume of the gas chamber in the suspension unit is known and that of the container fitted with the pressure gauge also is known, it is a simple matter to calculate the pressure of gas required in the latter container which, when discharged into the gas chamber of the suspension unit, will give the required pressure. So the operator can be instructed in the following manner. Clamp the suspension unit in its jig—this locks the crank arm in a position approaching the bump limit of its travel, so that the release valve is closed. Then, make all the necessary hydraulic and gas connections.

Next, turn off the cock serving the suspension unit. Open the cock between the gas cylinder and the container and, when the pressure on the gauge registers the appropriate amount, turn it off again. Should the pressure in the container be too high, a release cock can be eased open and then closed again when it has fallen to the required value. Next, the cock between the container and the suspension unit is opened so that the pressures in both are equalized. Finally, the pressure reading on the gauge is again checked, and adjusted to the required value by pumping oil into or bleeding it from the suspension unit.

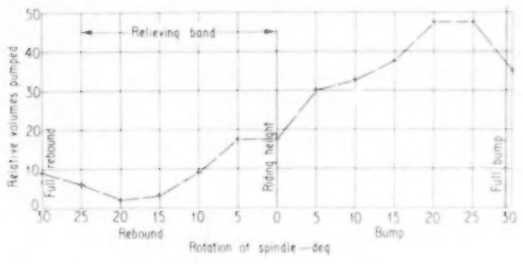
When the connections are initially made, a small plunger in each unseats a spring-loaded ball valve—these can be seen in the illustration on page 304. Then, when they are undone the valves seat again. As a further precaution against leakage during service, a plug is screwed into each hole and tightened hard down on a sealing washer. Finally, the ends of the plug holes are filled with a sealing compound, to discourage any subsequent unauthorized tampering.

Installation

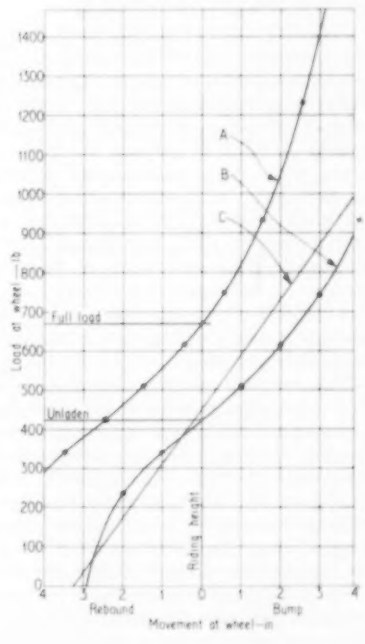
The Armstrong suspension unit is particularly suitable for cars in the 1½ to 2 litre class. Because the attitude in which the unit is installed is immaterial, its accommodation on a vehicle should present few difficulties. It can be applied as a conversion of an existing conventional suspen-

sion system, provided the structure can be suitably modified for its mounting. Obviously, though, the best results can be obtained in a vehicle specifically designed for it. This is not only because, with a specially designed installation, the best possible advantage can be obtained so far as use of space is concerned but also because, to obtain the optimum benefits in terms of softer suspension, the frequencies of the front and rear systems must be appropriately tuned relative to one another—normally, the frequency at the rear is, as is well known, the higher of the two.

The unit can be used as the suspension medium for all four wheels. Alternatively, it can be used for either the front or the rear pair. If it is applied at only one end, logically it should be at the rear. This is because the variation in loading on the rear wheels is greater than on the front ones of a car of conventional layout. Also, by using



Left: A plot showing the combined pumping and release characteristics for 5 deg increments of rotation of the spindle



Load deflection curves for the suspension unit, as applied to the rear axle of a 1½ litre car; these curves do not include the effects of the rubber buffers

A *laden characteristic;*
B *unladen characteristic;*
C *the curve for the original leaf spring*

it at the rear, the inherent disadvantages of the semi-elliptic leaf spring are obviated. However, axle location links are needed to take the place of the semi-elliptic spring. Several possible layouts are shown in the accompanying illustrations, and they include one in which a leaf spring is employed to locate the axle and to carry part of the load, while the Armstrong unit is used additionally to carry the remainder of the load and to effect the automatic levelling function.

In general, because the rate of the spring increases with the deflection, and because of the automatic levelling characteristic, the rate of the Armstrong unit can be appreciably less than that of a steel spring used for the same duty. Alternatively, smaller deflections can be allowed; and this, of course, has advantages in certain light commercial vehicle applications. An accompanying illustration compares the rate curves of the Armstrong suspension in the laden and unladen states with that of a leaf spring which was replaced by this unit in an experimental installation on a car.

Operation

The operation of the unit is fairly obvious from the illustrations showing its cross section and valves. Damping is effected by the large piston-and-cylinder assembly, while levelling is done by means of the pumping and release valves, the operation of which has already been described. One of the illustrations shows the relative volumes of hydraulic fluid pumped for increments of 5 deg of the crank rotation. From this it can be seen that the cam actuating the pumping valve is designed in such a manner that some pumping is occurring all the time, even when the release valve is open. In fact, the volumes pumped are small over most of the period when the release valve is open, and they begin to build up as the crank moves from the rebound to the static position. This means that as soon as the release valve is closed levelling begins immediately at a high rate. Since the suspension system does not sink overnight, as do some other systems, there is no urgent need for rapid levelling immediately it starts. Even if a heavy load is added, the condition as the vehicle moves off is not noticeably worse than it would be with a conventional steel spring installation.

No trouble has been experienced with regard to heat dissipation. During development tests, involving operation on pavé for protracted periods, oil temperatures have never risen above 60 deg C, although higher temperatures have been experienced on rig tests where the system is not subjected to the flow of air that normally occurs in vehicle installation. The reasons why the temperatures are so low are that the continuous circulation of oil within the unit helps, and so also does its large bulk, to dissipate and absorb heat. Moreover, the area of contact between the unit and its mounting is large and therefore there is ample scope for heat to be conducted away through the surrounding structure of the vehicle.

The first stage of development is now complete. That is, the unit has been thoroughly rig tested and proved in vehicle installations. The next stage is the further development of the unit to suit particular applications. Simultaneously, work can be done to refine the system, to reduce its cost and to adapt it for quantity production.

Cold Curing Silicone Rubber

A SILICONE rubber that cures without the application of heat, and is therefore very easy to use, is being manufactured by Midland Silicones Ltd, 68 Knightsbridge, London, S.W.1. It is called Cold-Cure Silastomer, and it is supplied in any of three consistencies: putty, paste and liquid. The addition of a small quantity of catalyst effects the curing action.

This product is particularly useful in experimental workshops. It can be used in applications such as small flexible moulds for prototype moulding, for insulating temporary wiring harnesses, for protecting small and delicate parts from shock, and for encapsulating electronic components. All of the characteristics of other silicone rubbers are claimed for this product — resistance to oxidation and moisture, and good temperature stability and dielectric properties.

Applications of

Some Details of Servo Brake System Installations

DURING recent years, a considerable demand has grown up for equipment intended to improve the overall braking performance in cases where cars and other vehicles are used for towing trailers, including caravans. This demand stems from the fact that, if the braking systems of the tractor vehicle and the trailer are not matched in terms of power and response, two dangers can arise : the first is inadequate retardation in an emergency, and the second is jack-knifing, when the trailer tends to over-run the tractor. However, in the conversion of existing vehicles to power braking, so great is the variety of layouts possible that the work is unsuitable for companies producing servo equipment in large quantities. It is therefore hardly surprising that a relatively small specialist company, Feeny and Johnson Ltd, of Wembley, Middlesex, has established itself firmly as one of the leaders in the conversional field.

Vacuum actuated trailer brakes

Probably the outstanding Feeny and Johnson product is the single-line trailer braking system, which has been in production for many years. The use of only one pipeline between the tractor and the trailer gives this layout a marked advantage in respect of simplicity. Normally the system is used to provide vacuum braking on a trailer towed by a vehicle, such as the Land-Rover, equipped with hydraulic brakes, but a version is available for use where the tractor has vacuum assisted brakes. The components of the system are in two groups, one on the tractor and the other on the trailer. In the first group are a source of vacuum, a dual-control hydraulic reaction valve and one half of a hose coupling, while in the other are the mating portion of the coupling and a vacuum power unit linked to the trailer brakes.

The reaction valve is, of course, the heart of the system. It is referred to as being of the dual-control type because it is actuated both by the hydraulic system for the brakes of the tractor vehicle and by an independent, cable operated mechanical control. The hydraulic connection to the valve is by way of a T-piece inserted in the brake line, and the mechanical control is applied by means of a hand lever situated near the driver, usually on the steering column, at a convenient distance below the wheel.

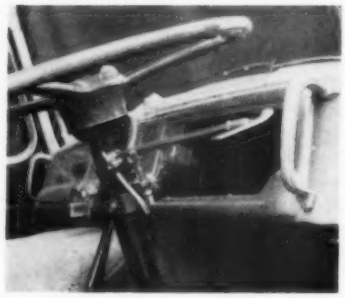
On the valve body there are two external vacuum connections, one of which leads to the source of vacuum—the inlet manifold in the case of a petrol engine, and an exhauster and vacuum reservoir where a diesel engine is fitted—and the other to the half-coupling on the tractor. Axially disposed within the body is a piston valve assembly, which is spring loaded towards the *brakes off* position. When the piston assembly is caused to move, by application of the brake pedal or lever, a circular flanged member mounted on it comes into contact with a rubber diaphragm, of annular form, clamped at its periphery between two portions of the body.

Further movement of the flange deflects the diaphragm off its seat, against the resistance of a second spring, and so permits the vacuum to be applied to the power unit. Progressive response of the valve to the control is obtained by applying the reaction from the power unit to a second diaphragm, thereby inducing a load that tends to reduce the opening of the main diaphragm. Thus, the latter quickly takes up an equilibrium position related to the effort applied to the brake control.

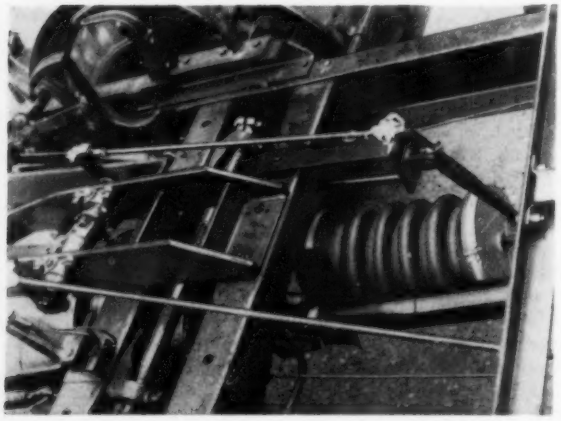
The coupling at the rear of the tractor is of the quick-release, self-sealing type comprising a part-conical male member, with a circumferential external groove, and a similarly formed female member carrying four captive steel balls, equally spaced in holes round the bore. In its engaged position, the male portion of the coupling is locked in position by the balls, which register in the groove, and its nose seats on a synthetic rubber sealing ring in the body. The final movement of engagement unseats a piston valve in the body of the coupling and so opens the circuit.

In this position, the balls are restrained against outward movement by the bore of a spring-loaded sleeve around the female member. Axial movement of the sleeve against the resistance of the return spring, however, brings an annular groove in its bore into line with the balls, which are thus permitted to spread radially outwards; the male portion can then be withdrawn. The withdrawal action, of course, allows the piston valve to return to its seat, without loss of vacuum.

Rubber hose of a composite type is used for all vacuum connections. The rubber is of the petrol and oil resistant type, and into it—near the periphery—is moulded a double insertion of open-weave canvas. As a safety measure, an additional valve of the break-away type can be mounted on the towbar. Should the trailer accidentally become detached from the tractor vehicle, the pull on the hose actuates this valve, thereby causing the full vacuum in an



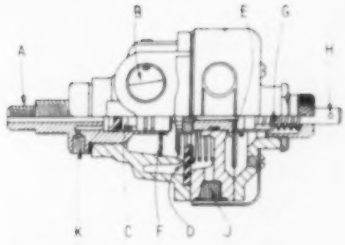
Right: In the Karrier Bantam conversion to power operated, trailer braking, a lever on the steering column provides a secondary control for the brakes



Below left: Detail view of the converted Bantam chassis. The convoluted servo unit actuates a lever that is connected to a second one on the Scammell slipper gear

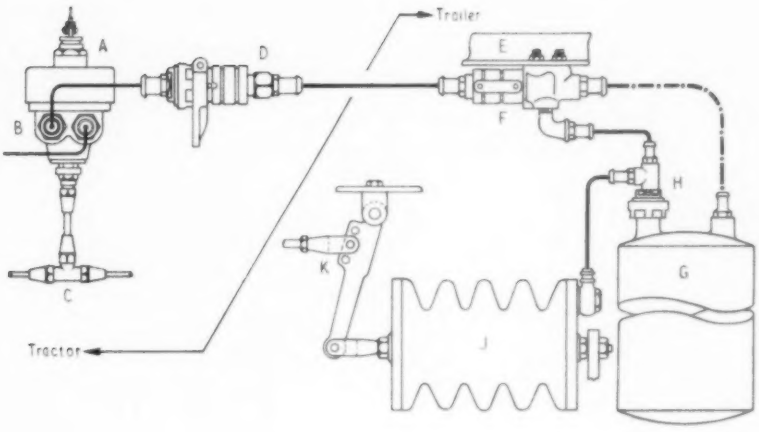
Vacuum Equipment

...d Other Work Carried Out by Feeny and Johnson Ltd.



A *connection to hydraulic system;* **B** *connection to vacuum power unit;* **C** *hydraulic piston;* **D** *main diaphragm;* **E** *reaction diaphragm;* **F** *filter gauze;* **G** *compensator;* **H** *hand control connection;* **J** *air release port;* **K** *bleed screw*

Above: Part-sectional view of the Feeny and Johnson dual-control reaction valve



A *dual-control reaction valve;* **B** *hose to source of vacuum;* **C** *T-piece in hydraulic pipeline;* **D** *quick-release coupling;* **E** *towbar of trailer;* **F** *break-away valve;* **G** *vacuum reservoir on trailer;* **H** *non-return valve;* **J** *vacuum power unit;* **K** *linkage to trailer brakes*

Diagrammatic layout of the single-line braking system for trailers towed by a vehicle with hydraulic brakes. The hose shown as a chain-dotted line comes into use only on operation of the break-away valve, should the trailer become detached

additional reservoir on the trailer to be supplied to the vacuum power unit, and so applying the trailer brakes.

Between the break-away valve and the reservoir are two pipelines, in one of which is a T-piece connected to the vacuum power unit. A simple non-return valve is fitted between the T-piece and the reservoir. The second pipeline forms a by-pass to the non-return valve and is opened to vacuum only in the emergency outlined in the previous paragraph, when the break-away valve operates. In this circumstance, the vacuum is applied to the power unit by way of the break-away valve and the T-piece.

To enable the installation to be suited to the weight of the trailer, five different sizes of power unit are made. These are extremely simple and are characterized by an absence of sliding friction and of any need of maintenance. Each consists merely of a rubber bellows, of convoluted form, and two cast aluminium end plates, to which the bellows is attached by circumferential clips, also of Feeny and Johnson manufacture. In each end plate is a central tapped hole. The hole in one plate is for mounting the unit, in this instance on a convenient point on the trailer; screwed into the hole in the other is an eye bolt, for attachment to the brake linkage. Because of the flexibility of the bellows, there is no need of a pivoted anchorage, as is required with a piston type servo unit.

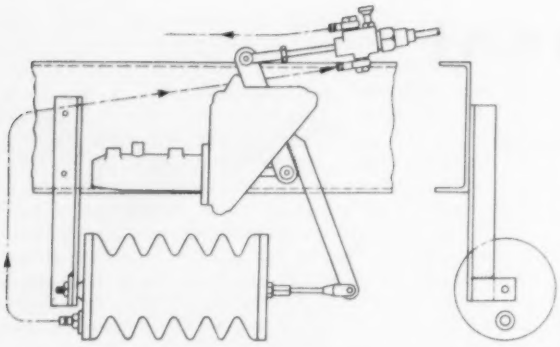
To prevent the convolutions of the bellows from collapsing radially under vacuum, a steel wire ring is moulded into the crest of each. In the development of these power units, which have now been marketed for over 30 years, the only real difficulty encountered was that of establishing the best mix for the rubber. Too hard a mix resulted in a bellows with an insensitive action, which required too much power to operate it, while with too soft a mix the bellows tended to bend in addition to collapsing axially.

An indication of the range covered by the five sizes of power unit is given by the following data. The smallest has an effective length, in its normally extended state, of $9\frac{1}{8}$ in and a diameter of 5 in, whereas the corresponding figures for the largest unit are $14\frac{3}{4}$ in and 8 in. At a vacuum of 20 in Hg, the minimum pulls exerted are 60 and 330 lb respectively, and the maximum strokes are $4\frac{3}{8}$ in and $7\frac{1}{4}$ in. There are three intermediate convolutions on the two smaller sizes, and four on the three larger models.

For use on diesel powered tractor vehicles, Feeny and Johnson Ltd. markets two exhausters, manufactured under Wade patents. These are the E.240 and E.300 models, the first of which has a maximum displacement of $3\frac{1}{2}$ ft^3/min, and the other of 5 ft^3/min. Both are of the eccentric vane type, with unguided vanes, and their maximum safe speeds for continuous operation are 2,950 r.p.m. and 1,100 r.p.m. respectively. Since each has an integral oil sump, no external source of lubrication is required. A V-belt drive is employed, usually from a pulley on the front end of the crankshaft; in certain applications, however, the driving pulley for the larger model is mounted on the propeller shaft.

A more specialized servo appplication, and one that has proved very popular, is to the trailer braking system of the Karrier Bantam tractor fitted with the Scammell J type coupling. As many readers are aware, this coupling is designed to provide rapid connection and disconnection of the trailer. Consequently, mechanical actuation of the brakes is employed, through the agency of a slipper mechanism at the rear of the tractor; this slipper gear engages automatically when the tractor is backed under the trailer for coupling purposes, and is disengaged by the action of the uncoupling control. The driver applies the trailer brakes by means of a separate lever in the cab, connected by a rod to an arm mounted on the shaft of the slipper unit.

In the Feeny and Johnson conversion, there are, in fact, three methods of operating the trailer brakes, the first being that just described, which is virtually unaltered. Both other methods involve the servo: the second is by way of the tractor's hydraulic braking system, and the third is by an independent manual control mounted on the steering column. A dual-control hydraulic reaction valve, of the type previously described, is therefore used, and its application is clearly shown in one of the accompanying illustrations. The power



This conversion of a normal hydraulic braking system to vacuum servo assistance has been fitted to certain E.R.F, Seddon and Vulcan chassis

unit is, of course, mounted on the tractor—not on the trailer as in the case of the system already mentioned—and sliding links are embodied in both the servo rod and the original control rod, to permit one to over-run the other.

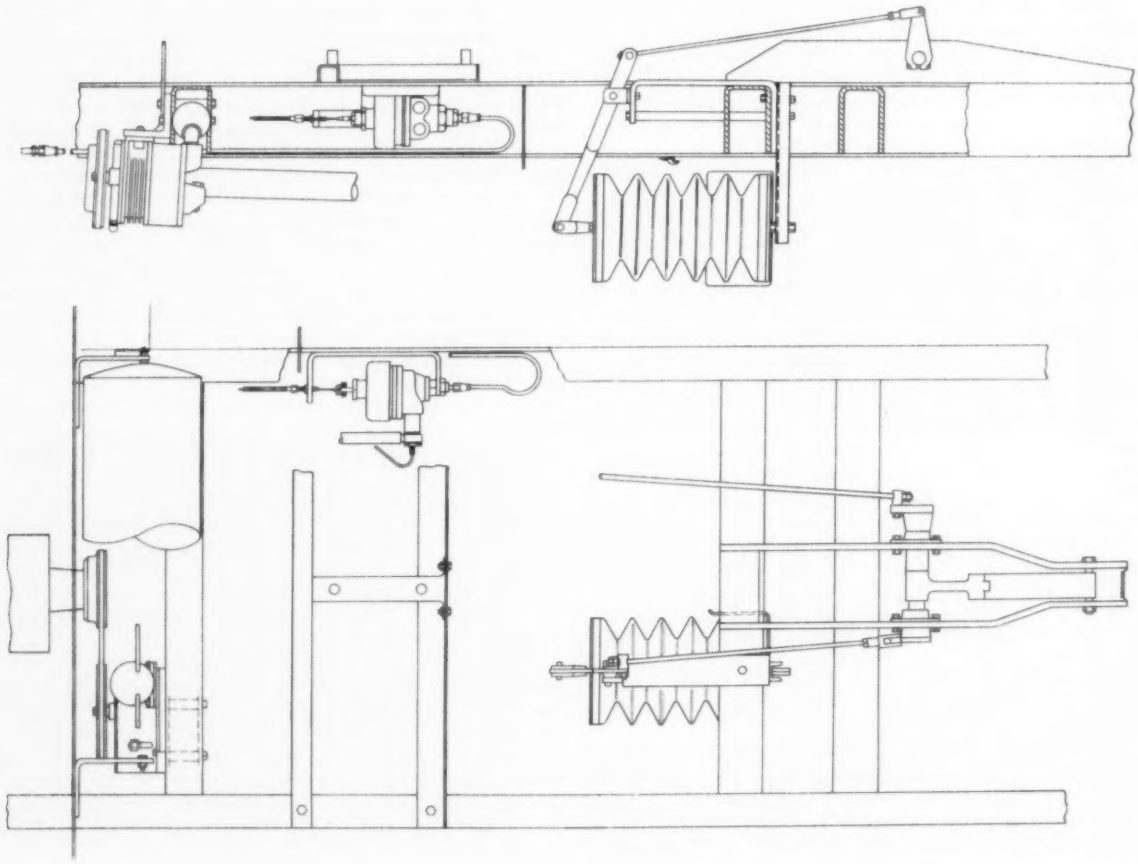
Servo-assisted braking

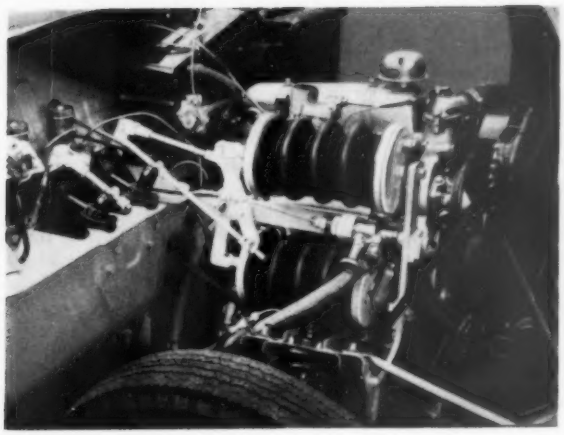
Another important activity of this company is the addition of vacuum servo assistance to vehicle and trailer brakes. A feature of this equipment is that any number of power units can be fitted in series for actuation from one control valve. The makers claim that their vacuum system does not suffer from significant time lag between depression of the pedal and response at the brakes. This rapid response is attributed to the design of the control valves and to the absence of pistons or other sources of friction in the power units. Control valves for mechanical as well as hydraulic layouts are available, so assistance can be provided equally well for hand or foot brakes. In all cases, the aim is at keeping the cost down by making the greatest possible use of existing parts and fitting the minimum of new ones.

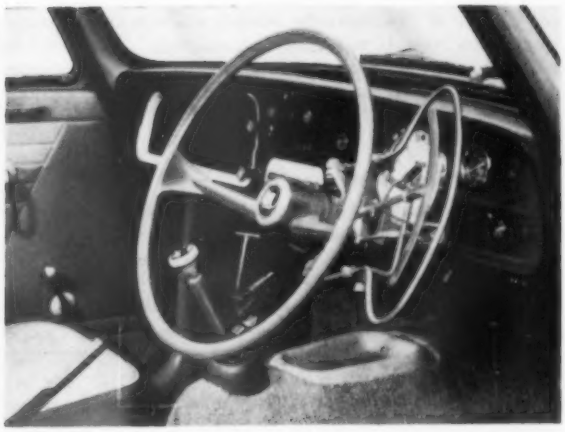
The most common conversion is to hydraulically actuated foot brake systems on commercial vehicles. It is usual for the control valve to be incorporated in the rod between the brake pedal and the lever actuating the master cylinder, whereas the power unit acts on that lever or an extension of it. To suit the individual layout, the control valve can be of either the tension or the compression type. The valve is not anchored to the chassis: one part of the pedal rod is screwed into the body of the valve and the other into its piston. A spring holds the piston in its neutral position when the brakes are off, the direction of the spring loading depending on whether the valve acts in tension or in compression.

In addition to actuating the master cylinder, the initial movement of the brake pedal moves the valve bodily and closes its breather, which carries a small air filter element. Further movement results in increasing resistance at the brakes; when this resistance becomes sufficient to overcome the valve spring, the piston begins to move and applies some

Reproduction of the general arrangement drawing showing the conversion of the Karrier Bantam tractor fitted with the Scammell J type coupling







Two views of a Triumph Herald car converted for a legless driver. On the left can be seen the vacuum power units and their linkage to the pendant pedals for the clutch and brakes. The clutch control is carried on the gear lever and those for the brakes and throttle are on the steering column

vacuum to the power unit to assist the pedal movement. The reaction from the power unit opposes the movement of the piston, so that a lap position is reached as soon as the pedal load attains a steady value. An increase in pedal load, of course, causes the piston to move further, thus increasing the degree of vacuum assistance. By this means, a fully progressive action of the valve is obtained.

A drawing of a typical application of servo assistance, suitable for certain E.R.F, Seddon and Vulcan chassis, is reproduced. In this instance, the control valve is of the tension type, and the existing rocking lever for the master cylinder is adapted by welding on an extension that provides the correct leverage for the power unit. In the case of a vehicle with a petrol engine, the only additional parts necessary for this simple installation are the valve, the power unit and its mounting bracket, and the hoses. A diesel model would, of course, have to be fitted with an exhauster and a vacuum reservoir.

Hand control conversions

A large proportion of Feeny and Johnson's capacity is given over to the manufacture of conversion kits that enable cars and commercial vehicles to be driven by people who have lost the use of one or both legs. For the convenience of customers, the installation is normally carried out not by the company but by an agent, of whom there are eight in Britain and twelve in other parts of the world. It is of interest that about 40 per cent of the output of this equipment is exported, Israel being one of the main importers. Systems have been evolved for most of the popular cars, including the B.M.C. ADO 15, the Triumph Herald and the Ford range, as well as for various Rover models and other more costly vehicles.

Since the hand control equipment is additional to, and does not replace, the existing foot controls, a converted car can still be driven normally, and can readily be restored to its original condition for resale; in view of the limited market for vehicles with hand control, this second point is one of some importance. It is probable, too, that much of the equipment could be used again for converting the succeeding car. In all cases, the throttle is hand operated, and so also, where neither leg can be used, are the brake and clutch. For owners having a left-leg disability, the clutch is converted for manual actuation, as is the brake where the right leg is inoperative.

The following description covers the layout used in the full conversion. Basically, the accelerator and brake controls are grouped together on the steering column, below the wheel, and the clutch control is mounted on the gear lever. Servo actuation is applied to the brake and clutch controls but not to the throttle control, because the effort needed is obviously insufficient to warrant the additional complication. Bowden cables are employed for all three controls and, to ensure the minimum of friction over a long period, the inner wires are treated with a dry lubricant—actually a graphited wax—on assembly. This wax is preferred to oil or grease, because it does not tend either to pick up dirt or to dry out under the influence of engine heat.

The throttle control cable is taken from the finger lever directly to the carburettor, if the existing control is by cable, or to a convenient point on the linkage in the case of rod operation. To permit the driver to turn the steering through a reasonable angle without loss of contact with the lever, this is of T shape, with an arcuate bar. Manoeuvring in confined spaces and restarting on hills are facilitated by the operation of a small secondary lever that locks the main lever in any desired position. If required, this facility can, of course, be used to give a fixed throttle opening for cruising on clear roads.

As can be seen from the illustration, the brake hand lever is of T-shape, formed from a continuous length of steel rod in such a way that it circumscribes the throttle lever, with sufficient clearance to permit easy operation of either lever independently of the other. Simultaneous actuation is possible but this requires an above-average level of manual dexterity. Various clutch control levers have been designed, and the type used in a particular application depends primarily on whether the gear lever is of the floor mounted or steering column type; consideration is given to the length and angle of a floor mounted gear lever, to ensure that the clutch lever is conveniently to hand for gear changing.

In the case of both the clutch and the brake controls, the actuating cable is connected to a progressive vacuum control valve of the type already described. This valve applies the requisite degree of vacuum from a reservoir to the appropriate bellows unit, the moving end of which is attached to the end of a relay lever. In turn, this lever is coupled to the pedal by means of a sliding link, which permits the pedal to be used without bringing the servo into action.

Because of the wide differences between the layouts of various cars, it is clear that no standard installation would be practicable, so each new application has to be developed empirically. In the typical installation illustrated, both servo units and their control valves are fitted under the bonnet,

but because of considerations relating to space available and accessibility, this is not always practicable. The vacuum reservoir can, of course, be stowed in any convenient position, the most common being the boot. A vacuum gauge, for mounting on the facia panel, is included in all conversion kits.

Piston type servo units

Finally, brief mention should be made of the fact that the company manufactures a second range of servo units, of the piston and cylinder type. These are available with bores of between 2½ in and 10 in, and there are alternative strokes for each bore size. A feature of all these units is their lightness, which results primarily from the use of plastics cylinders. In the case of the smaller sizes, the end caps are aluminium castings, but light-gauge steel spinnings are used on the larger models. The caps are secured by a series of long bolts, the purpose of which is to minimize the stresses in the cylinder.

Two sizes of these servo units are at present in regular production: one has equal bore and stroke measurements of 8 in, and for the other these dimensions are 2½ in and 1⅝ in respectively. The larger model is employed on a mechanical road sweeper, for raising and lowering the brush gear, while the smaller one is for adjusting the suspension damper settings on the Vanden Plas Princess 4 litre car.

Vibration Meter

TO meet the demand for a compact and portable yet very accurate and versatile instrument for the measurement of vibration, Dawe Instruments Ltd, of 99 Uxbridge Road, London, W.5, have developed the type 1431 vibration meter. This unit is fully transistorized and gives direct readings of velocity, displacement and acceleration over a wide range. By virtue of its inherent freedom from microphony and the absence of any warming-up period—resulting from the use of transistors—this instrument should be well suited for work on vehicles and machinery.

The meter can be operated from normal a.c. mains. Alternatively, power can be provided from dry batteries accommodated within it. The power consumption is only 2W and therefore eight small torch batteries are sufficient for 80 hours of operation.

A moving coil pick-up, whose output is directly proportional to the velocity, is used to sense the vibration. Integrating and differentiating networks can be switched into the circuit to give direct readings of displacement and acceleration. An output jack is provided so that the amplified signals can be displayed on a cathode ray oscillograph or can be fed to a frequency analyser or other types of equipment.

The pick-up has a resonant frequency of approximately 5 c/sec, which is well outside the frequency range of the meter, from 10 to 1,000 c/sec. Its vibration range is such that a peak-to-peak maximum amplitude of approximately 0·6 in between stops is covered, and the maximum peak acceleration that can be measured is 10,000 in/sec^2, or just under 26g. Despite this high top limit, the sensitivity in the lowest range is such that a displacement, from zero to peak, of only 0·0003 in gives full-scale deflection, and readings can be made directly to 0·000005 in. The plan dimensions of the instrument are 7 in × 7½ in, and it is 10½ in high. Without batteries, the unit weighs approximately 6½ lb and complete with batteries its weight is about 7½ lb.

Apprentice Executives

BECAUSE of the ever increasing need for highly trained commercial executives in its widening organization, Leyland Motors is to introduce a commercial apprenticeship scheme. The aim, in keeping with Leyland tradition of promoting from within the company, is at providing a wide and varied training in the latest business and management techniques, to fit the trainees for senior executive positions with the Leyland Group companies at home and overseas.

Applicants at the age of 16 should preferably have passed the G.C.E. at O level in four subjects including English and mathematics. Those who are accepted by the selection committee will start work in September.

They will first go through an intensive training period in most departments of the commercial division. From this they will gain practical experience in the costing, internal audit, material control, purchasing, sales contracts, shipping, statistics, wages and many other departments. After this, individual arrangements will be made for these apprentices to specialize in the type of work for which their natural aptitude and inclination makes them most suitable. This will continue till they finish their apprenticeship at the age of 21.

Throughout their first year, the apprentices will be released from work for one day each week, with pay, to attend a preparatory commercial course in the company's own school. They will also attend for one evening a week to learn a foreign language. Later, additional day release facilities will enable the boys to take the Ordinary National Certificate in commerce at local technical colleges. On successful completion, suitable apprentices will be given a further day release to obtain a more specialized professional qualification of the recognized accountancy or secretarial associations or institutes. Further details can be obtained on request from Leyland Motors Ltd, Leyland, Lancs.

Computer Staff

AN ANNOUNCEMENT has been made to the effect that Senior Staff Consultants Ltd. have formed a special department to deal exclusively with the recruitment and appointment of staff for all aspects of computer work. To assist them in this project they have retained the services of Computer Consultants Ltd. and have taken over the specialist staff appointment service previously run by that company. Enquiries should be addressed to Senior Staff Consultants Ltd, 7 Cork Street, London, W.1.

Heat Bibliography

A BIBLIOGRAPHY, containing over 4,000 references, has been prepared by the National Engineering Laboratory. It is entitled "Heat Bibliography 1959" and includes references to papers, reports and books on subjects covering heat exchangers, basic heat transfer processes, the physical properties of gases and liquids, instruments for measuring temperature and flow rate, the economics of heat exchange plant, the effects of corrosion and fouling on the efficiency of heat exchangers, and many other associated topics.

The information was obtained from original publications and abstract journals, and all relevant titles noted in NEL during 1959 have been listed. A simple subject classification arrangement has been used to avoid extensive cross referencing. This work will be of particular use to manufacturers and users of all types of heat exchange plant, including boilers, condensers, liquid coolers, refrigerators, and distillation equipment, and to research workers in the field of heat transfer. It costs 20s, postage 1s 9d extra, and is available from H.M.S.O, 13a Castle Street, Edinburgh.

New Plant and Tools

Recent Interesting Developments in Production Equipment

CONTINUOUS bright heat-treatment, brazing and annealing can be carried out in the Hump-Back conveyor furnace manufactured by Royce Electric Furnaces Ltd, Albert Drive, Sheerwater, Woking, Surrey. There are three chambers in this furnace: pre-heating, heating and cooling; they are joined by gas-tight connections and together form a continuous metallic muffle, so that hydrogen or cracked ammonia can be used as protective atmospheres. The endless conveyor belt is of nickel-chromium wire, riding over the base of the muffle in the furnace, and returning on skid plates beneath the furnace. A continuously rated, squirrel cage motor transmits drive through a variable speed unit to the large, rubberized roller that drives the conveyor belt.

Two basically similar furnaces are in the Hump-Back range, the "W" type having heavy gauge nickel-chromium wire heating elements, for temperatures up to 1,000 deg C, and the "T" type nickel chromium tape elements for temperatures up to 1,150 deg C. The elements of the "W" type furnace can be removed individually through the side walls, whereas the "T" type has a removable roof, which can be raised for access to the furnace chamber. Both pre-heating and cooling furnaces are water-jacketed, but the main chamber is lined with high-grade refractory tiles and semi-refractory brickwork.

These furnaces are available in four sizes, the widths of the conveyor belts being respectively 4 in, 6 in, 8 in and 12 in. The overall length of the longest of the furnaces is 34 feet, and that of the shortest 16 feet. In addition to the standard furnaces of the range, the manufacturers can supply units to meet special requirements.

On this tube-bending machine, the order of a succession of bends is determined by pins on a rotating drum that is behind the front cover
(Chamberlain Industries Ltd)

Programmed tube-bending

An interesting addition to their range of tube-bending machines has been introduced by Chamberlain Industries Ltd, Staffa Works, Argall Avenue, London, E.10. This machine, a semi-automatic unit with a pre-selector device that determines the direction and sweep of up to twelve successive bends, is manufactured by the Herber Company of Sweden. In this country it will be known as the Staffa-Herber type SBA 28.

The central feature of this machine is a horizontal turntable, on which are two forming tools, the adjacent faces of which act as the jaws of a vice during a bending operation. When a tube is to be bent, it is gripped by the tools and wrapped around one of them as the table rotates; the bend can be either right or left-hand, according to the direction of rotation that is selected for the operation of the turntable.

Conveyor furnace, with gas-tight muffle for inert atmospheres, for bright heat-treatment or annealing parts up to 12 in diameter
(Royce Electric Furnaces Ltd)

Around the turntable, but fixed to the body of the machine, is a graduated ring, on which are clamped blocks in which are incorporated pneumatic plungers. These blocks can be placed in any desired angular position. Where a selected plunger is set to protrude, it is contacted by a stop on the rotating turntable, and the turntable is halted and automatically returns to the starting position.

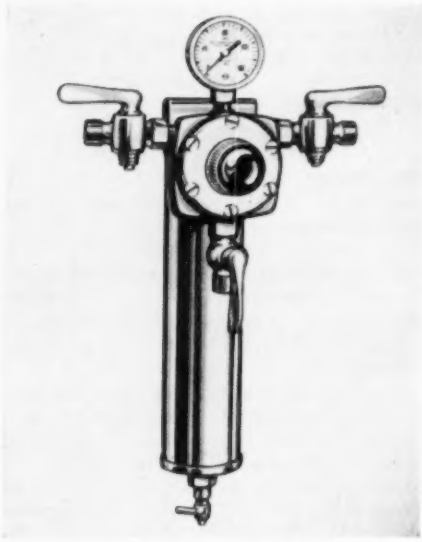
The valves that control the plungers are mounted in a row, and are operated by steel pins screwed into a rotating preselector drum. Two of the pins select the direction of rotation of the turntable, and the remainder are placed in an order appropriate to the order and sweep of the bends, as decided by the angular positions of the relevant stops on the turntable ring. At the start of a bending operation, the tube is fed from the front of the machine, through the forming tools and on to a long mandrel. Each bend is started by the depression of a foot-rail, and as it is completed, the tube is automatically fed forward to its next position, determined by a series of pre-set ratchet-type stops on the workholder.

Bends in different planes can be accomplished by resting the tubes on special jigs fixed to the turntable and set at the appropriate angles. It will be appreciated that on this machine, all bends must normally be of the same radius, but it is possible to have two tools of different radii and to have all left-hand bends of one dimension, and all right-hands bends of another.

The operating drum, then, stores the programmes for the order of bends, and the sweep of each bend has to be adjusted by hand. It is possible to store more than one programme on a drum, provided the total number of bends does not exceed twelve. If any untoward incident occurs during the execution of a programme, the drum can be returned to zero by pressing a button; it is thus not necessary to complete the programme in order to start bending the next piece of tube that is due to be fed into the machine.

The flexibility of this abrasive sponge is well illustrated. With it, adjacent faces of differing profiles can be treated in one pass
(Finishing Aids and Tools Ltd)

Below: A device for separating oil and water from compressed air and for reducing the magnitude of the fluctuations in air pressure
(Manufacturer, Aerograph-DeVilbiss Ltd)



This machine is 38 in high, and has a capacity of $1\frac{1}{8}$ in o.d. × $\frac{1}{16}$ in wall thickness steel tube, or $1\frac{1}{8}$ in × $\frac{1}{16}$ in brass tube; the minimum radius of bend is $1\frac{1}{8}$ in, and maximum 6 in. The length of the standard workholding mandrel is 10 ft, and the length of tube automatically fed forward allows the forming of bends up to a maximum of 150 deg.

Abrasive sponge

Because of the flexibility of the Artifex abrasive sponge the claim is made that it is possible to reduce the number of operations required to give a high finish to a wide variety of materials. Close and continuous contact is maintained between sponge and workpiece, and only light pressures are necessary. Adjacent faces that are curved, flat and profiled can be finished with one pass, and the inherent absence of chatter is said to impart a finish of a consistent standard over the treated area.

No coolants, lubricants or polishing compounds are required with this material, but if the need arises to use them, the effectiveness of the sponge is unaltered. The Artifex sponge is manufactured by Finishing Aids and Tools Ltd, Chandos House, Buckingham Gate, London, S.W.1, and is available in four grades: soft, medium hard, hard and extra hard. Matt or polished surfaces can be produced on all ferrous and non-ferrous metals, and on plastics and hardwoods. Grinding wheels are supplied in various sizes up to 16 in diameter and 4 in thick.

Air transformer

Designed to provide a steady, unfluctuating supply of clean air to all types of air tools, the type HLE air transformer has recently been announced by Aerograph-DeVilbiss Co. Ltd, 47 Holborn Viaduct, London, E.C.1. This device is particularly useful when used in conjunction with paint-spraying equipment, for it can provide two large spray guns with a constant supply of air at any pressure likely to be required. Within the body of the transformer there is an air-driven centrifuge for separating oil and water from the air supplied, and also a filter for removing small particles that are likely to cause damage to the centrifuge.

Delivery pressure can be controlled to within $\frac{1}{4}$ lb/in^2 for each 10 lb/in^2 of fluctuation in supply pressure from the mains. This transformer has a flow capacity of 50 ft^3/min, and the maximum delivery pressure is 135 lb/in^2, the highest usable supply pressure being 250 lb/in^2. It is basically a tall cylinder, with the regulating device and pipe connections at the top, and its overall height is quoted as being 15 in.

Gas Carburizing Plant

Wild-Barfield Semi-automatic, Sealed-quench Equipment Installed at the Standard-Triumph International Ltd. plant

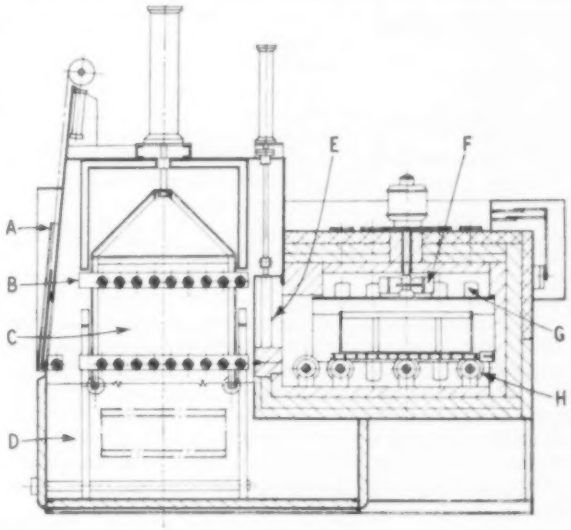
APART from economic considerations, the most important advantage of gas carburizing is that it enables precise metallurgical control of the quality and the depth of case to be exercised. This installation at the Fletchamstead works of Standard-Triumph International Ltd, comprises four Wild-Barfield ACE single-zone, sealed-quench furnaces and two horizontal, forced air-circulation, tempering furnaces, for gas carburizing and hardening and for tempering respectively. They are employed for heat-treating transmission gears and components. So far as the gas carburizing equipment is concerned, the prime requirement was that the quality and carburizing potential of the atmosphere used should be consistent and should not require supervision by the furnace operator. The carburized components were to be so treated that no final post-treatment cleaning or grinding would be necessary.

One of the drawbacks in the conventional fixed-retort, pit-type, gas carburizing furnace is that the charge is exposed to air between furnace and quench, and consequently a slight scaling results. For most production items this can be tolerated but, where a scale-free finish is required, an alternative type of equipment is desirable. Hence, sealed-quench furnaces were chosen for this particular installation. They are versatile in application and can be used either for gas carburizing or carbo-nitriding, as well as for carbon restoration, clean-hardening, and clean-normalizing. Their main characteristic is that from the time a charge is placed in the furnace chamber until its removal after cooling or quenching, it does not come into contact with air.

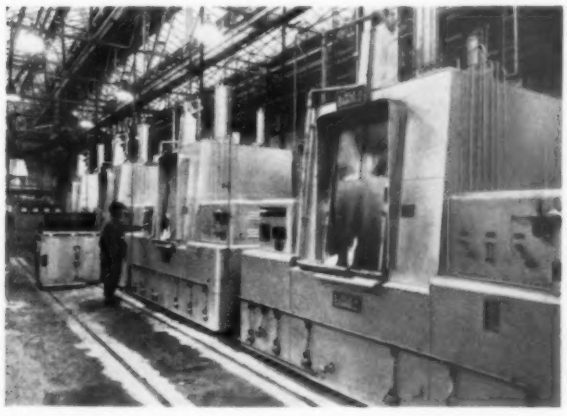
The diagrammatic section gives a general impression of the equipment which, as may be seen, is virtually divided into two sections. On the left is the entry chamber and on the right the carburizing section, with a gas-tight door between them. The entry chamber contains a loading platform that can be raised or lowered. When lowered for quenching, there is a period of oscillation in a vertical plane and, simultaneously, a violent agitation of oil through the charge. This ensures that work will be free from soft spots

A *outer door*; **B** *double tier load and quench platform*; **C** *entry chamber*; **D** *quench tank*; **E** *inner door*; **F** *circulating fan*; **G** *radiant tube heating elements*; **H** *roller hearth*

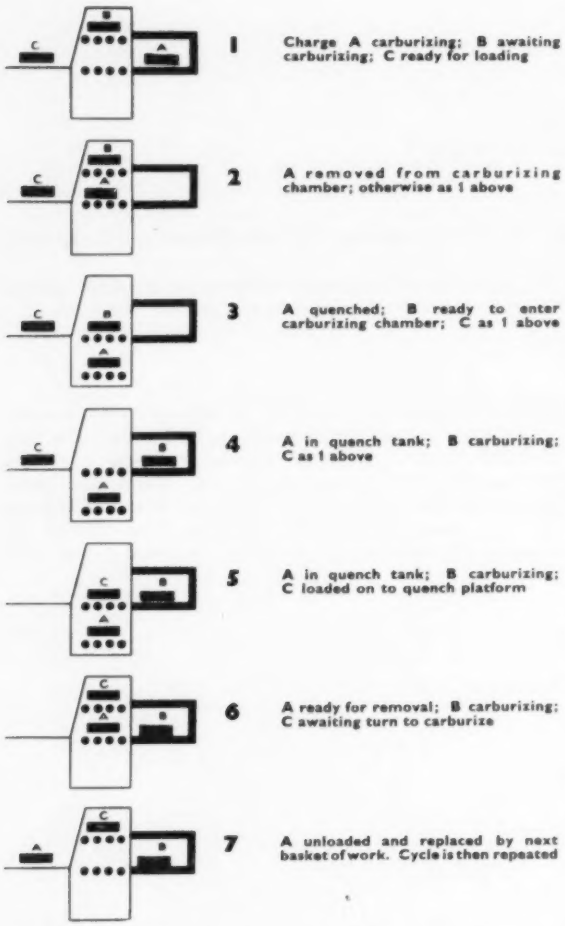
Diagrammatic section of Wild-Barfield ACE sealed-quench furnace



Battery of sealed-quench furnaces at Standard-Triumph International



caused by small pockets of quench-oil vapour trapped in the work. The quench tank, which forms part of the entry section, is provided with oil circulators, coolers and immersion heaters; the quenchant temperature is maintained at a pre-set figure by thermostatic control. Quench equipment has been so designed that, when required, hot-oil quenching can be carried out. To facilitate the insertion and withdrawal of the work baskets containing components undergoing treatment, the main carburizing chamber is fitted with a roller hearth. If desired, the hearth roller-assembly can readily be modified for power operation, thus easing the work of the operator. A fan is provided in the roof of the carburizing chamber to ensure an effective circulation of atmosphere through the charge, in which duty it is assisted by the provision of a suitable baffle. Heating is provided by Wild-Barfield electric radiant-tube elements,



Operational cycle for carburizing followed by direct quenching

fully protected against excess temperature conditions and earth leakage and completely sealed from the chamber atmosphere. These units are of a design that permits speedy replacement to be made, if necessary, whilst the furnace is in operation and without interfering with the atmosphere conditions. The provision of a double-tiered platform in the loading section enables semi-continuous production to be maintained. The sequence of operations is shown in the diagram.

Atmosphere control

Of the atmospheres available for gas carburizing, the one chosen for use in this installation is endo-propane enriched with propane. The advantages of this atmosphere stem from its low sulphur content, purity, and consistency of properties. To produce a really satisfactory carburizing gas, it is necessary to have a specific "carbon potential" relating to the steel, the case depth required, and the carburizing temperature. Generally speaking, this "carbon potential" is related to the carbon dioxide and water vapour constituents of the atmosphere. Both of these constituents are present in fractional amounts and, of the two, the water vapour is by far the easier to measure with accuracy.

It has become customary to measure and express the water vapour content as the dewpoint of the atmosphere, and this value can be co-related to the carburizing capacity of the atmosphere under stated conditions. This means that by varying the dewpoint it is possible to control the surface carbon content of a steel, and during carburizing to control both this value and the carbon gradient of the case. Although several methods exist for the control of dewpoint, by far the best solution is found in fully automatic control; that is, by a recorder which activates automatic correction of gas quality to a pre-set dewpoint. A system of this type can hold the control to ± 1 deg F dewpoint although in practice, for many operations, a coarser control of up to ± 3 deg F is satisfactory. The method is in use with the Wild-Barfield ACE furnaces at Standard-Triumph and enables consistent gas carburizing to be obtained with the minimum of supervision by specialized personnel.

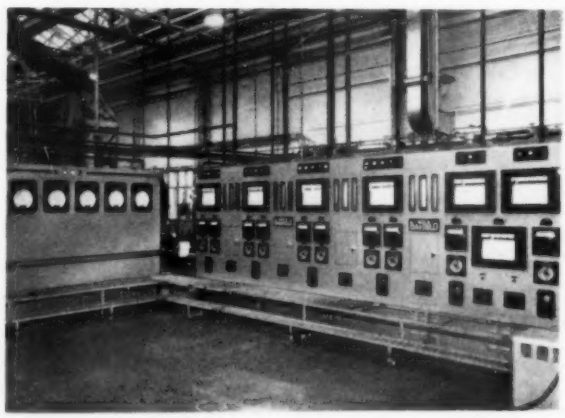
Operating conditions

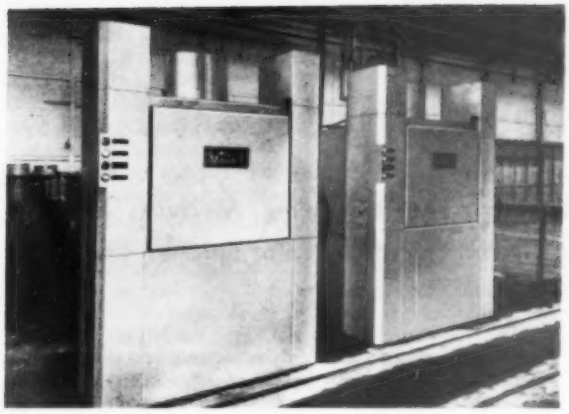
At Fletchamstead, the charge load of components—gear shafts, gear wheels, cluster gears, differential pinions and similar parts—is approximately 400 lb. Carburizing is carried out at 925 deg C for 1·5 hr, followed by a stabilizing period of 2·5 hr for diffusion and eventual reduction of temperature to 815 deg C. The parts are then quenched in oil which is maintained at a temperature of 53-54 deg C. In normal operation the overall treatment time is from 5·5 to 6·0 hr, which includes 1·0 hr recovery time. The effective case depth is from 0·025 in to 0·030 in.

Operations are controlled by four sets of instruments on a large panel. A Honeywell electronic strip chart recorder-controller operates in conjunction with two process timers which automatically reset on completion of the cycle. Each set of instrumentation includes manual controls for the propane and air producing the endothermic gas. There are also Honeywell Pyr-O-Vane instruments that indicate and cut off fuel flow under excess temperature conditions, and similar instruments safeguard the remotely sited gas generator. A separate recorder-controller keeps the quench oil temperature constant at the pre-set value to within a tolerance of $\pm 1{\cdot}0$ deg C. Dewpoint control is provided by Foxboro-Yoxall circular chart controllers.

In addition to the ACE furnaces, the installation includes two special tempering furnaces. They are situated at the end of the line of ACE equipments, remote from the control panel. Each is of normal box-type construction with coiled heating elements of nickel-chromium alloy. A baffle inside the chamber, in conjunction with a suitable fan, ensures a positive air circulation which is conducive to rapid heating and temperature uniformity. The pilot lights seen in the illustration to the left of each door are repeat signals from the furnace control panel and indicate "mains on", "furnace

Furnace control panels, with dewpoint controls mounted on the left





Two special tempering furnaces for the Standard-Triumph installation

on" and "excess temperature", while the fourth gives a visual signal that the treatment period has expired. The last-mentioned light operates jointly with an audible signal. Honeywell controllers for these two furnaces are mounted on the main instrument panel. It should be added that a degreasing unit is provided between the ACE furnaces and the tempering equipment.

The emphasis in these notes has been upon gas carburizing but, as was pointed out earlier, the ACE furnaces can be used for carbo-nitriding. In practice, the carbo-nitriding is carried out at 160-180 deg C using a normal carburizing atmosphere to which a small addition of ammonia has been made. The process is, of course, similar to gas carburizing in that both temperature and time are related to case depth.

Seal Quality Control Equipment

ABOUT four years ago the General Motors Research Laboratories, of Warren, Michigan, U.S.A, embarked on a research programme concerning the causes of leakage through lip type oil seals. This programme was initiated because leaking seals were undesirably frequent at that time; in many cases, of course, rectification involved extensive dismantling and reassembly, the cost of which was far greater than that of the seal. The data obtained during the programme indicated that accurate testing equipment for seals of this type was essential if an adequately high standard of reliability was to be maintained. Two types of machine were therefore designed by the Laboratories and are now manufactured by them.

These two machines, designated Sealrater and Sealector, are complementary, and the following outline of the various factors affecting the performance of a seal will explain why both are necessary. In the first place, the seal must be tight enough only to stop leakage, and not so tight that it squeezes out the protective oil film under the lip. An excessively tight seal gives rise to high local temperatures, with consequently rapid wear or even burning of the lip, either of which leads to early leakage. The tightness, clearly, can be caused by either too small a diameter in relation to the shaft, or too high a lip pressure, which in turn can derive from either excessive spring loading or an over-hard rubber mix.

Eccentricity between the seal and the shaft must be closely controlled because of its modification of the effects of both the diameter and the lip pressure. The shaft, too, plays an important part in minimizing the risk of leakage. On this component, a surface finish of between 10 and 20 microinches is regarded by the Laboratories as the best compromise: it is rough enough to support the oil film under the lip, yet smooth enough not to break through that film and cause direct contact between shaft and seal.

In addition, if the shaft has a turned finish, the machining lead should be less than the width of the working face of the seal. If it is not, rotation of the shaft in the appropriate direction will result in oil being pumped past the seal. The shaft, too, must be round enough not to cause a standing gap to form in the lip at high rotational speeds. Finally, care in storing the shaft, and in the subsequent assembly, is necessary to avoid the nicks and scratches that can nullify the care taken in other directions.

The conditions under which an oil seal operates can obviously vary considerably from one application to another. It follows that the correct tolerances for one application will probably not suit another. The function of the Sealrater machine, therefore, is the determination of these tolerances by testing under an accurate simulation of the appropriate operating conditions. Not only can the running speed of the shaft be varied as required, but any desired start-stop cycle can be applied; other variables include oil temperature and pressure, and the axial travel of the seal on the shaft. Oil seeping past the seal under test is absorbed by felt washers, which are weighed before and after the run to ascertain the degree of leakage. In this way, the effect of a whole range of seals differing in respect of diameter and lip pressure can quickly be investigated under controlled conditions, and the necessary tolerances determined.

Because of the haphazard nature of production variations, no system of statistical sampling can eliminate the potentially faulty seal, so a 100 per cent check is regarded as essential. The Sealector machine was designed to carry out such checks quickly and efficiently, against the tolerances set by the Sealrater. In its fully automatic form, the Sealector can check up to 1,200 seals per hour for lip diameter and pressure. The principle of the machine is simple: a batch of seals is fitted on to a series of stub shafts, and the flow of compressed air between the seals and the shafts is gauged. A measure of the lip diameter is given by the rate of flow at a certain air pressure; the lip pressure is indicated by the air pressure needed to produce a given flow rate. Semi-automatic and laboratory versions of the Sealector are also available from the Laboratories.

Western Australia

A REPORT by a team of nine British industrialists who visited Western Australia in October 1960, at the invitation of the State Government, has just been published. The objectives of the visit were to investigate opportunities to establish new industrial capacity and generally to stimulate in the United Kingdom wider interest in the development of secondary industries in Western Australia, which is 2,000 miles nearer to Europe and the expanding markets of Asia and Africa than are some of the Eastern States. This booklet can be obtained from the Office of the Agent General for Western Australia, whose address in the United Kingdom is Savoy House, 115 Strand, London, W.C.2.

Grinding Ball-Tracked Components

Internal and External Thread-grinding Machines Specially Developed for the Quantity Production of Steering Gear Parts

WIDER use in various industries of recirculating ball devices for translating rotary motion into linear movement has raised problems in the production of ball tracks in the mating components. In the automobile industry, specifically for steering gear elements, to the essential precision of machining is added the need for rapid production in quantity. In the manner characteristic of the machine tool industry, the Coventry Gauge and Tool Co. of Fletchamstead Highway, Coventry, has developed two machines specially to meet these requirements.

Internal grinding machine, TI 1315

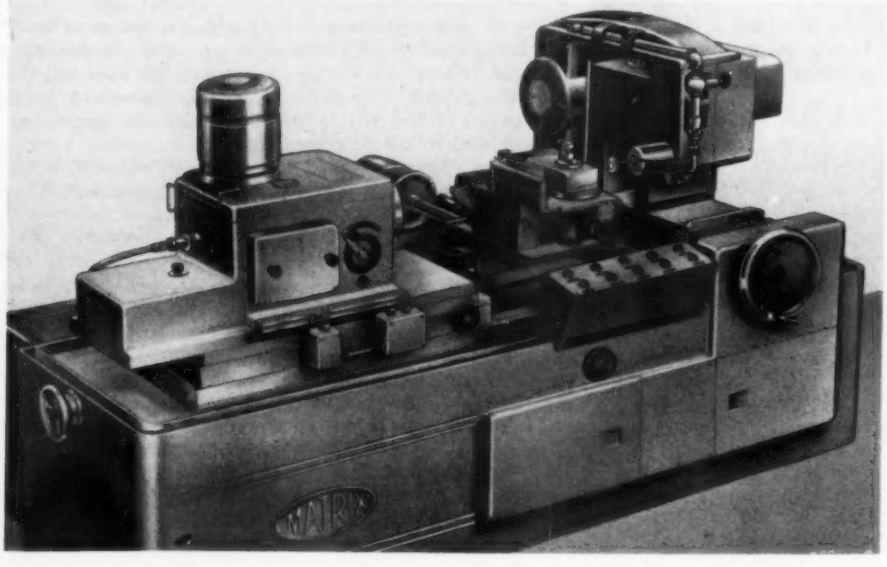
Intended for a wide range of application, the Matrix model TI 1315 internal thread-grinding machine for ball-tracked nuts and half-nuts has a maximum capacity of work up to 13 in diameter and can grind threads up to 10 in diameter. Apart from the loading and unloading of workpieces the machine is fully automatic in operation. A single ribbed grinding wheel is used, being automatically dressed by a diamond at predetermined intervals. Either single-pass or multi-pass grinding routines can be carried out; the maximum number of passes is twenty-four and the total amount of automatic in-feed of the wheel is 0·25 in.

On full nuts the grinding action is arranged to be continuous, the wheel in-feed taking place at each reversal of the work slide. When grinding half-nuts, however, a different method is employed in order to reduce the time that otherwise would be expended in "grinding air". The wheel feeds into a stationary workpiece to the pre-set incremental depth, the work spindle commences rotation and the work slide traverses. At the end of the traverse, work rotation ceases, the wheel is retracted to clear the work, the work slide returns to its starting position, and the cycle repeats. To facilitate unloading and reloading, the wheelhead is automatically retracted to its rear position when a workpiece is finish ground.

Two types of grinding wheel spindles are selectively available; one belt-driven and the other high-frequency. For the belt-driven version a 5 h.p. motor having speeds of 15,000 and 18,000 r.p.m. is fitted. The high-frequency drive is steplessly variable from 12,500 to 40,000 r.p.m. Maximum diameter of the grinding wheel is 3 in. A swivelling movement is provided for the wheelhead unit to enable right-hand or left-hand threads with helix angles up to 15 deg to be ground. As standard, the maximum thread length that can be ground is 5 in. By means of a special wheelhead design this length can be increased to 7 in, but it entails a reduction of the maximum helix angle to 10 deg.

The rapid withdrawal movement of the wheelhead to the loading and dressing position is pneumatically actuated, as also is the automatic compensation for wheel dressing. A special, diamond, dressing unit is mounted on a swivelling cradle in order that the wheel can be formed accurately while the wheelhead is inclined at the helix angle of the thread being ground. Wheel dressing operations can be carried out before or after the final, finish-grinding pass.

Workhead drive is by a 1·5-3·0 h.p. motor providing seven rotational speeds from 6 to 105 r.p.m. or 12 to 210 r.p.m.



Matrix model TI 1315 thread grinding machine for ball-tracked nuts and half-nuts

Return speed is the same as the forward grinding speed, but to reduce idle time when grinding at slow forward speeds, a speed of 56 r.p.m. is available. The pitch of the thread being ground is determined by interchangeable leader bars and nuts. Each bar can be used for either right-hand or left-hand threads of identical pitch.

Maximum distance from the spindle nose to the grinding wheel centre is 15·5 in. The workhead is mounted on a sub-slide so that it can be positioned longitudinally, relative to the grinding wheel, to facilitate the grinding of short-length threads in long components. For the production of multi-start threads, the spindle is provided with an indexing mechanism to position it for two, three, four, five, or six starts. Other features of the workhead are an adjustment to control parallelism and a side-cut control to enable a pre-roughed thread to be aligned with the grinding wheel.

Coolant is delivered through the workhead spindle, which has a 2 in diameter bore, and also by an external pipe to the grinding wheel. Another delivery pipe is fitted at the wheel dressing position. To prevent coolant splash from the machine, and to ensure efficient use is made of the fume extractor provided as standard equipment, the upper half of the machine is enclosed by a manually retractable hood.

External grinding machine, TW 712

Matrix external thread-grinding machine, model TW 712, has been specially designed and developed for the precision grinding at high speed of ball-tracked steering worms and similar items. Work to a length of 12 in can be accommodated between centres and the maximum thread length that can be ground at one setting is 8 in. The recommended maximum thread diameter is 3 in and, of course, both right-hand and left-hand threads can be produced. Maximum swing on the work centres is 5 in.

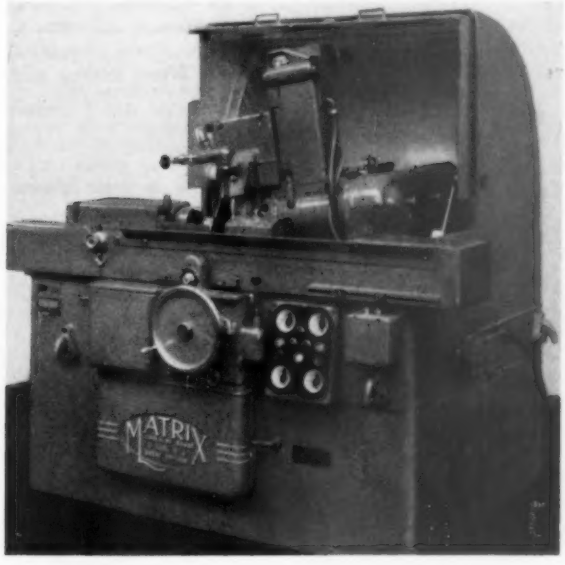
Grinding wheels of 18 in, 16 in, or 14 in diameter can be used, running up to a maximum peripheral speed of 10,000 ft/min. The wheel spindle runs on angular contact ball bearings and the drive is by vee belts from a dial-wound 3·5 h.p. motor providing grinding and dressing speeds. Pulley changes in the belt drive can effect speed reductions of 10 per cent and 25 per cent from maximum speed. The wheelhead can be swivelled, by worm and wormwheel, up to 15 deg on each side of the vertical for grinding right-hand or left-hand helices. Automatic wheel dressing, and also compensation for wheel dressing, is included in the machine cycle. Dressing can be arranged to take place after a preselected number of components has been ground and the diamond can be set to remove 0·0005 in to 0·003 in from the wheel at each dressing.

The workhead is driven by an 0·6 h.p. motor, a range of spindle speeds from 3 to 30 r.p.m. being obtained through pick-off gears. Traverse of the work slide is by a leader bar and nut which is readily interchangeable with others for work of different pitch. Length of traverse is determined by adjustable trip stops and micro-switches. A former bar can be mounted at the front of the machine to impart a slight cross movement to the wheelhead, thus barrelling the component should this be desired. As on the internal grinding machine, a side-cut control is provided for aligning pre-machined work with the grinding wheel.

A normal grinding cycle is as follows:

1. rapid advance of wheel to the grinding position
2. work spindle rotates and component is traversed past the wheel
3. wheel advances by feed increment and work spindle and table reverse
4. work spindle stops and wheel retracts to rest position for work unloading and reloading.

Depth of cut for the finish grinding pass can be adjusted from 0·0002 in to 0·008 in as required to suit work in hand.



Above: Grinding a ball-track helix on a steering gear worm shaft
Left: The external thread grinder can handle work up to 10 in long

Lubrication is effected by a dual pump unit supplying oil under pressure to the various bearings through regulating needle valves. Other parts are either flood- or oil-bath lubricated and all vital points are provided with sight glasses.

The coolant tank and pump are arranged as a free-standing unit on castors to facilitate servicing and cleaning. A coolant separator is available as extra equipment. During grinding operations the wheelhead and table are totally enclosed, the raising and lowering of the hood being included in the machine automatic cycle. A fume extractor unit is supplied as standard. Vaporized coolant and grinding fumes are drawn from under the hood by an extractor fan and passed through a series of filters before being expelled. Electrical contactors and overload switches are housed in a separate cabinet and electrical controls are grouped on the machine casting. In the United Kingdom the sole selling agent for both these machines is the Rockwell Machine Tool Co. Ltd, Welsh Harp, Edgware Road, London, N.W.2.

Flame Hardening of Camshafts

By Dr. Ing. H. W. GRÖNEGRESS

Considerations of Component Design and Burner Arrangement for Rapid and Economical Production by the Spin-hardening Method

FLAME hardening owes its widespread industrial application to the fact that it provides a means of accurately localizing the hardening treatment and confining it to those surface areas of a component which are subject to wear. The use of high capacity burners working on a gas-oxygen mixture enables the surface undergoing treatment to be supplied with far more heat per unit of time than can be dissipated into the core by thermal conduction. There is thus a build-up of heat in a thin surface zone, and as a result the component, after quenching, acquires a thin hardened layer on the surface of a relatively soft and tough core.

This method of heat treatment automatically effects a substantial saving in heat energy and time compared with those required for conventional case hardening. In addition, since the core remains cold, it offers resistance to deformation, so that any hardening distortion which may arise is markedly reduced. As a consequence, grinding times can be cut; even if not, as in many cases, eliminated altogether. Hardening times are in close agreement with machining times, and this means that the hardening machine can be integrated in the actual production line. The hardening process itself is clean and does not give rise to any dirt or dust, so that high-grade machine tools adjacent to the hardening equipment do not suffer in any way. However, there are certain factors affecting both the design and material aspects of the work to be hardened which must be borne in mind if full benefit is to be derived from the advantages of flame hardening. The purpose of the following remarks is to examine these factors.

Fig. 1. Examples of desirable and undesirable features of design

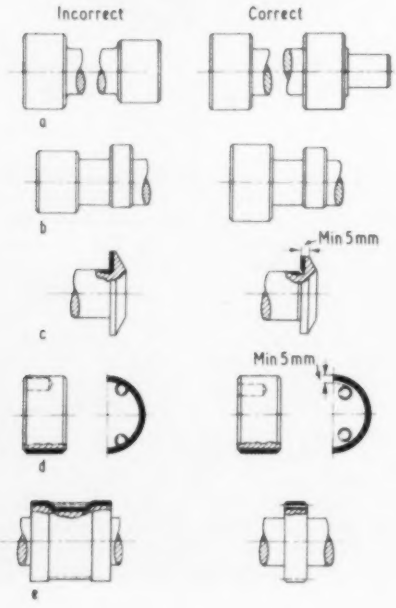
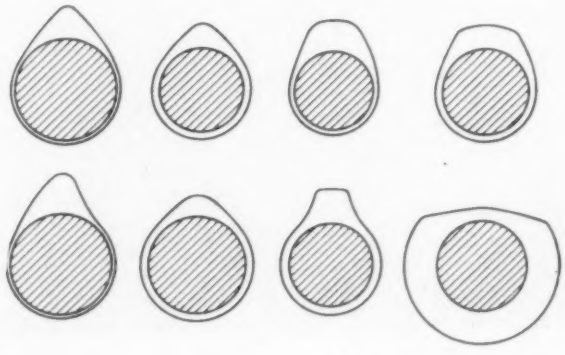


Fig. 3. Polished sections of hardened cams. Left, incorrect treatment; heating too intense. Right, carburized case on steeply flanked cam

Design considerations

As an example of the technical possibilities of flame hardening it is proposed to consider the flame hardening of camshafts for internal combustion engines and fuel injection pumps. Hardening is carried out by the spin method; in other words, the component is rotated and the cams and journal diameters to be hardened are first brought up to hardening temperature and then quenched. In contrast with the individual, successive heating of the cams and journal diameters, as employed in the induction hardening method, flame hardening enables all journals and cams to be heated simultaneously. This feature allows substantial cuts to be made in production times. When dealing with

Fig. 2. Typical range of cam shapes. All can be flame hardened



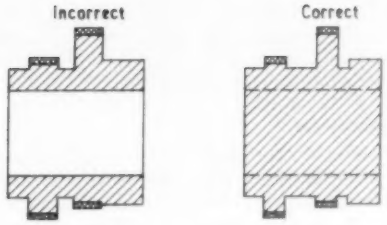


Fig. 4. Incorrect camshaft design can result in non-uniform hardening of contours

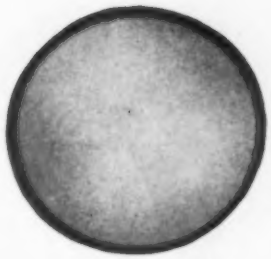


Fig. 5. Polished section of a hardened camshaft journal

very long camshafts it sometimes becomes desirable to divide the hardening operation into two or more stages. For this purpose, the hardening machine is equipped with a patented magnetic-valve control system enabling the various stages of the process to run successively without necessitating removal of the work from the machine or reloading.

For the spin method the heating time depends on the diameter of the work and can be taken as approximately 0·5 to 0·7 second per mm diameter for the sizes of camshaft under consideration. From this it follows that all the journals should preferably have the same diameter, as in Fig. 1(a). Furthermore, the diameter of the journals should be somewhat larger than the tip-circle diameter of the cams, see Fig. 1(b), since this enables automatic feeding of camshafts in automated hardening plant to be accomplished without the use of complicated handling gear. In cases where the end bearing is furnished with a flange for accurate axial location of the camshaft, it is possible to harden the thrust face of this flange at the same time, but it is desirable to separate the two adjoining faces by an undercut fillet to simplify the hardening process. The preferred method is indicated in Fig 1(c).

It is important to avoid locating holes immediately under the surface to be hardened, as at Fig. 1(d), since in such positions they tend to promote through-hardening and, possibly, hardening cracks. Camshaft pinions can be hardened by the spin method in the same operation without any difficulty; the teeth being through-hardened down to about 1 mm above the root circle. The best arrangement is

Fig. 6. Section of flame-hardened camshaft in a silicon-chromium-molybdenum alloy cast iron

for the pinion to be formed individually, as illustrated at Fig. 1(e), and not to cut it in one of the spaces between two bearing diameters, since the resulting unequal distribution of metal makes it difficult to obtain uniform heating. The surfaces to be hardened must be machined to a good standard of finish. In the case of cams it is necessary for the side faces also to be left free of grooving if spalling of the cam noses is to be avoided. Where the cams are of a complex shape or made of a material prone to this type of damage, it is usually of advantage to provide a slight radius on the edges of the cams.

Depending on the rotational speed and working conditions of the engine concerned, so may the cam noses be of more or less pointed design. Typical contours are shown

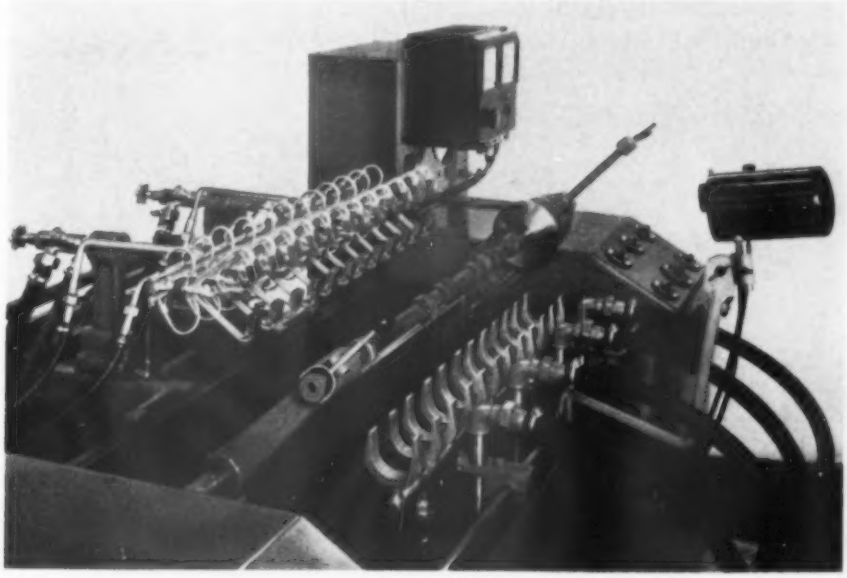


Fig. 7. Semi-automatic hardening machine U 103, with special burners for a 4-cylinder camshaft, and the Milliscope temperature-sensing, control device

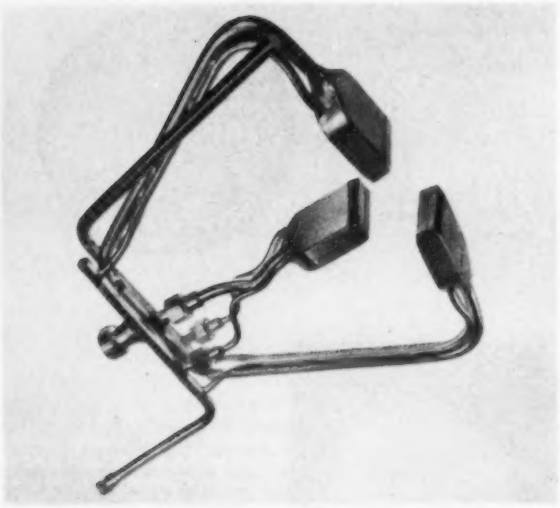


Fig. 8. Three-jet burner unit for hardening a single camshaft journal

in Fig. 2. Fuel injection pumps are characterized by the fairly broad shape of their cams. Pointed cams, however readily tend to overheat at the tips and, therefore, in such cases the distance between the burners and the work should be reduced so that the nose of the cam passes through the core of the flame. When very steep cam profiles are involved, the trailing ramp tends to be deprived of heat so that the depth of hardening undergone by it is shallower. This is particularly the case when the material used has pronounced surface hardening properties, as shown in Fig. 3 (left). Difficulties of this nature can, however, be overcome by reducing burner power and possibly by splitting up the total heat output over a number of burner jets.

Finally, it is important to ensure proper and balanced distribution of metal. Fig. 4 (left) shows that the base of the cam is practically identical in diameter with the shaft. The result of this is that heat is conducted away very rapidly at that zone and, consequently, it is difficult to obtain a uniform depth of hardening. The correct practice, therefore, is to step the cam up from the shaft all round, as shown on the right of Fig. 4. Furthermore, the axial boring should be done after completion of flame hardening. This presents no difficulty, since the heat treatment is limited to a shallow surface zone, as is well illustrated in the section of a camshaft journal, Fig. 5.

Choice of material

Such a short heating period as 0·5 to 0·7 second per mm diameter does not allow any chemical change to take place in the material. In other words, the surface zone is neither carburized nor decarburized. This means that the material used must already possess the carbon content needed for it to take on the required hardness. The carbon content of the material governs the degree of surface hardness attainable, whilst the depth of hardening can be influenced by the alloy content. For camshafts in general, a hardening depth of from 2 to 3 mm is sufficient, and in such cases the camshafts can be made of plain carbon steel. A satisfactory grade for this purpose is Ck 45 (equivalent EM 8) having 0·45 per cent carbon and yielding a Rockwell hardness of about 58 ± 2. If a minimum hardness of 60 Rockwell is specified, a steel having a carbon content of 0·53 to 0·60 per cent should be used. These grades, however, are prone to cracking and require quenching in emulsion or oil. In such cases, the hardening machines must be provided with the necessary equipment for circulating and cooling the quenching medium.

Steel camshafts are produced by drop forging, calling for expensive forging dies which do not justify the cost involved when only short runs are envisaged. For this reason there have been many attempts to make camshafts in cast iron. When casting against chills, however, it was found to be extremely difficult to ensure that all the cams take on a uniform degree of hardness since the cams are, of course, disposed in various planes. Chill casting also rules out the possibility of re-hardening, and this leads to a relatively high reject rate in manufacture. This explains why industry

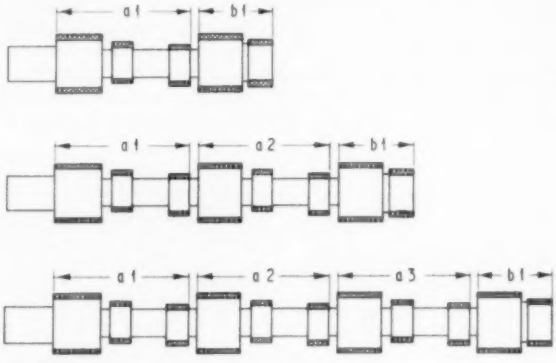


Fig. 9. Camshafts with standardized grouping of cams and journals. Only two different burner assemblies are thus required

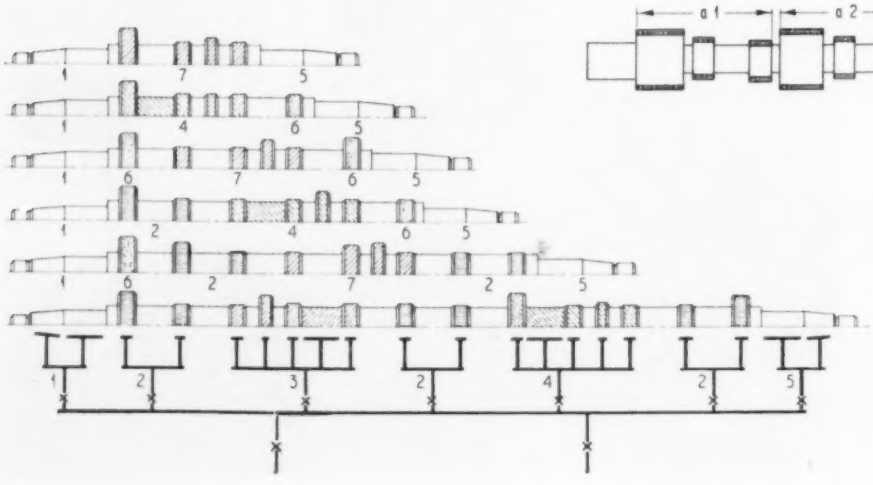


Fig. 10. Diagram showing the use of standardized cam and journal group burner assemblies for hardening a full range of camshafts

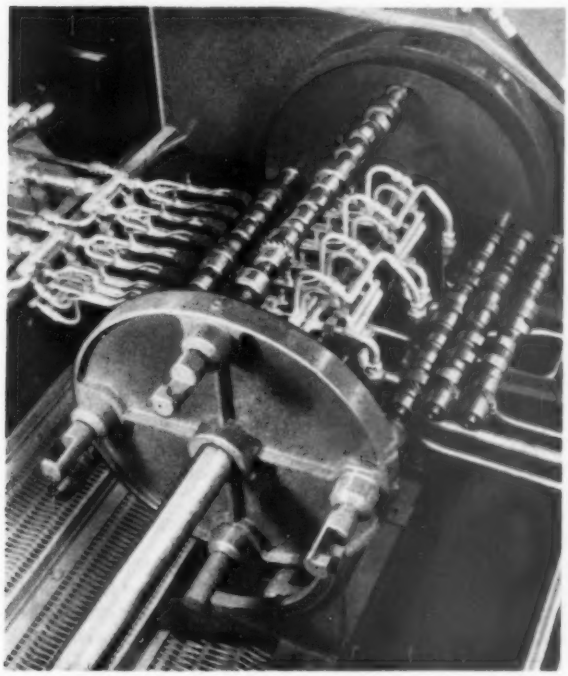
is giving up the practice of chill casting camshafts and changing over, to an ever increasing extent, to surface hardening after casting.

Modern casting methods allow the required shape to be imparted to a camshaft with very great accuracy. It becomes possible, therefore, to dispense with the machining of cast camshafts prior to hardening, and all that is necessary after hardening is finish-grinding. This is conditional, of course, on the use of alloy cast iron yielding a depth of hardening of at least 3 to 4 mm. Alloy cast iron of this grade has given excellent performance under practical vehicle operating conditions, see Fig. 6. Owing to the slower cooling rate of alloy cast iron it is necessary in this case also to quench in emulsion. A typical alloy composition for this purpose is C—3·5, Si—2·5, Mn—0·8, P—0·073, Cr—0·85, and Mo—0·33 per cent.

When it is important that the depth of hardening should follow the cam contour faithfully, particularly when steep cam profiles are involved, it may be preferable to make the camshafts of case-hardening steel. The carburizing process is then appreciably simplified since there is no need for masking as protection against the carburization of those areas which are to remain soft. With flame hardening it is possible to localize and control the hardening process with accuracy and apply it only to those areas subject to wear. It should be emphasized, on the other hand, that this practice makes it possible to improve the strength of the end bearings—normally protected against the penetration of carbon by masking during carburization—by hardening them simultaneously with the hardening of the cams. In this way an improvement in the torsional strength of the shaft is obtained.

For the volume production of camshafts it is convenient to use specially designed burner assemblies, as in Fig. 7, for each camshaft. In such cases, two or three water-cooled burners, Fig. 8, are assigned to each journal and cam and act in unison on the surface to be hardened. The surface undergoing hardening is thus surrounded by an envelope of flame which brings about a rapid rise to hardening temperature in the surface zone and permits efficient utilization of the heat energy supplied. Such burners necessarily have to be made by hand because their lifespan is so long that the demand for them is small. They are, therefore, relatively costly and their main application is in installations where flame hardening is carried out in the actual production line, using hardening machines specially adapted for a specific type of camshaft.

When camshafts are required in a variety of shapes and sizes and in comparatively small numbers it is generally better to take the hardening machine out of the production line and put it in a central hardening shop. In a case like this, the designer can make a useful reduction in the number of different burners required if he applies standardization principles to the dimensions and pitches of the cams. Standardization of camshafts on these lines is a fairly easy matter where new designs are concerned. Its introduction is only possible stage-by-stage when an existing



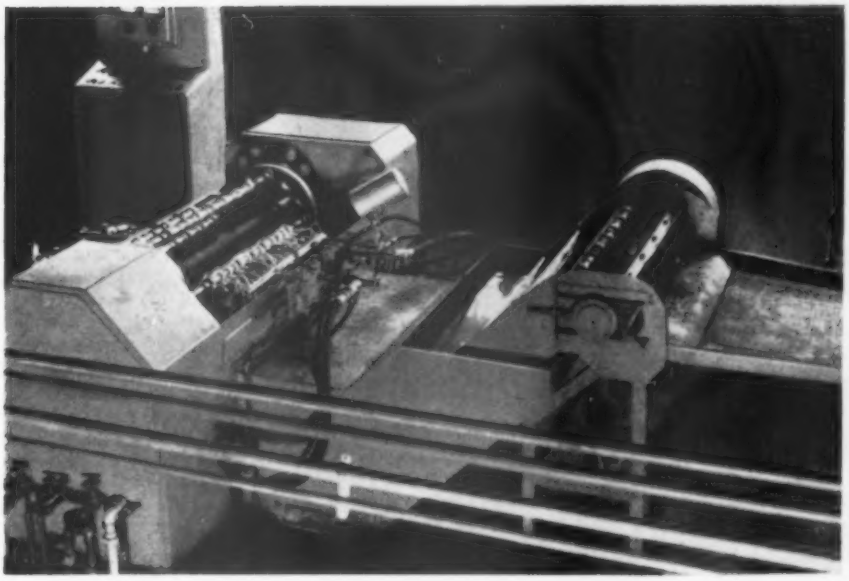


Fig. 12. Burner station on the NWA fully automatic hardening machine, showing cam device for unloading work at quench station

Fig. 11. Automatic spin-hardening machine NWA installed in a production line in a car plant

production set-up has to be changed over, for example, from a conventional case-hardening method to flame hardening.

In such circumstances, therefore, it is necessary to ascertain whether there are not at least certain groups of cams and bearings having the same dimensions and pitches so that special burners can be made up for these groups. These burners can then be connected together by distribution pipes to form a complete burner assembly for hardening a given type of camshaft. Fig. 10 shows a typical case in which a number of different camshafts exhibit definite groups of cams and bearings which recur at the same pitches and in the same dimensions. To these groups are assigned burners connected to a common distributor line and with the aid of such groupings the burner assemblies needed for hardening various types of camshaft can be built up. This method has proved to be thoroughly practical as a means of reducing the initial outlay, particularly when short runs are involved.

Selection of the hardening machine itself will depend on the quantities to be handled. In the simplest machine, the model U 103 shown in Fig. 7, the camshaft to be hardened is mounted in a pivoting carrier. The carrier is swung upwards to bring the camshaft into the burners and downwards to bring it under the quenching spray. The same pivoting action automatically controls the operation of the burners and of the quenching spray by means of magnetic valves, so that the attendant has only to operate the control

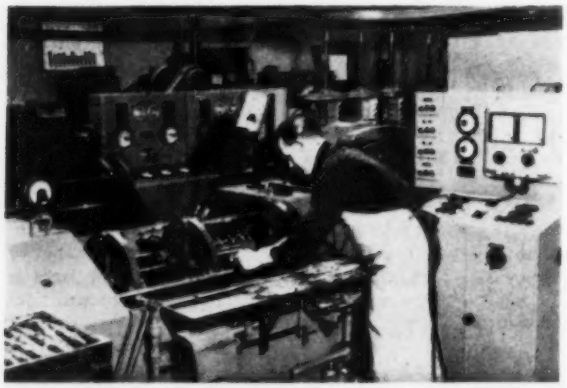


Fig. 13. NWA automatic machine equipped with two work stations. Control console is on the right and two flow-metering cabinets at rear

lever for heating, for quenching, and finally to unload the finished camshaft and reload the carrier with another for treatment.

Heating is controlled by the Milliscope*, a temperature measuring device specially developed for flame hardening and operating with zero time lag without requiring any physical contact with the work. These instruments can be seen mounted in position in Figs. 7 and 11. As soon as the Milliscope shows that hardening temperature has been reached the operator must lower the heated camshaft into the quench spray. The cooling period is terminated by a timing relay on which the required quenching time has been pre-set.

The pivoting action can also be made automatic with the aid of air cylinders, the duration of the heating being controlled either on the basis of hardening temperature, by

* "*Automobile Engineer*", *October* 1955.

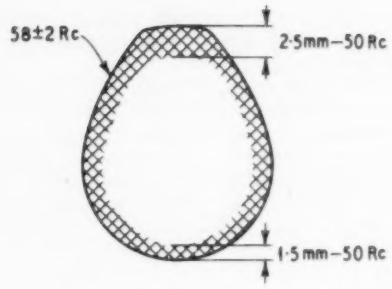


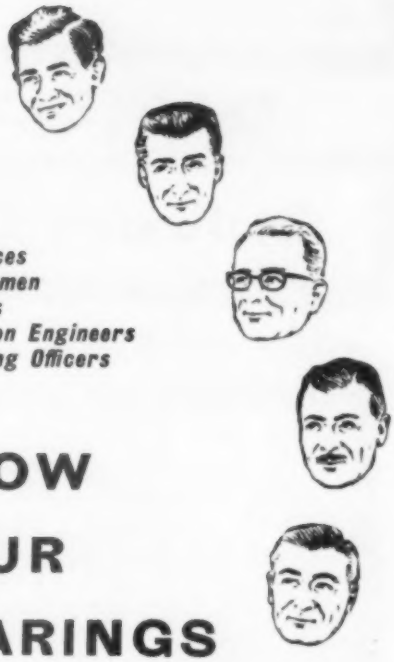
Fig. 14. Hardness pattern and data for typical automobile engine cams. Gas pressure 0·3 atm, consumption 43 m³/hr; oxygen pressure 8·5 atm, consumption 24 m³/hr; heating time 36 sec; quenching time 28 sec

using the Milliscope, or on a time basis by using a built-in timing relay.

Taking the quenching time as approximately equal to heating time, and allowing time for unloading and reloading the machine, it can be reckoned that a machine of the type described is capable of hardening about 50 to 60 two-cylinder or four-cylinder camshafts per hour. When handling larger camshafts, six-cylinder units for example, the hardening operation, as mentioned earlier, is divided into two stages with the aid of a patented magnetic valve control system. Even when operating in this manner it is still possible to harden about 25 to 30 camshafts per hour. When dealing with the smaller types of camshaft capable of being hardened in a single operation, it is sometimes possible to increase the output of the machine by arranging for the heated camshaft to be automatically released after introduction into the spray. It is then deposited on roller stands which effect continued rotation of the camshaft under the spray so that another camshaft can be loaded on to the machine carrier during the cooling phase, and the heating processs recommenced.

When the numbers involved are larger, and particularly when only one specific type of shaft is to be treated, the model NWA automatic camshaft hardening machine gives excellent results. The carrier indexes to four positions, the first being a loading station, and at the second drive is applied to the camshaft. Heating takes place at the third position and quenching at the fourth. The work transport utilizes a Geneva-type mechanism. Camshafts can be fed in from a magazine and automatically clamped, provided that the bearing diameters are larger than the tip-circle diameter of the cams. Unloading can also be made automatic at the quenching station. The camshaft then drops on to a conveyor belt and is taken to the hardness-testing machine or passed on for finish machining. Fig. 11 shows a general view of the machine and Fig. 12 illustrates the burner layout. For handling small camshafts this automatic machine can be furnished with two stations, as in Fig. 13, enabling up to 200 camshafts to be hardened per hour.

Should the work to be hardened require quenching in emulsion or oil, the automatic camshaft hardening machines are equipped with a heat exchanger and with a pump for circulating the quenching medium. The temperature of the quenching medium is checked by a built-in thermostat which also controls the performance of the heat exchanger to ensure that there is no unnecessary consumption of cooling water. For the sake of obtaining consistent results it is advisable to use a flow-metering unit when making the burner settings. The values established by the metering unit and proved by testing are entered in the component drawings so that reproducibility of results is guaranteed.



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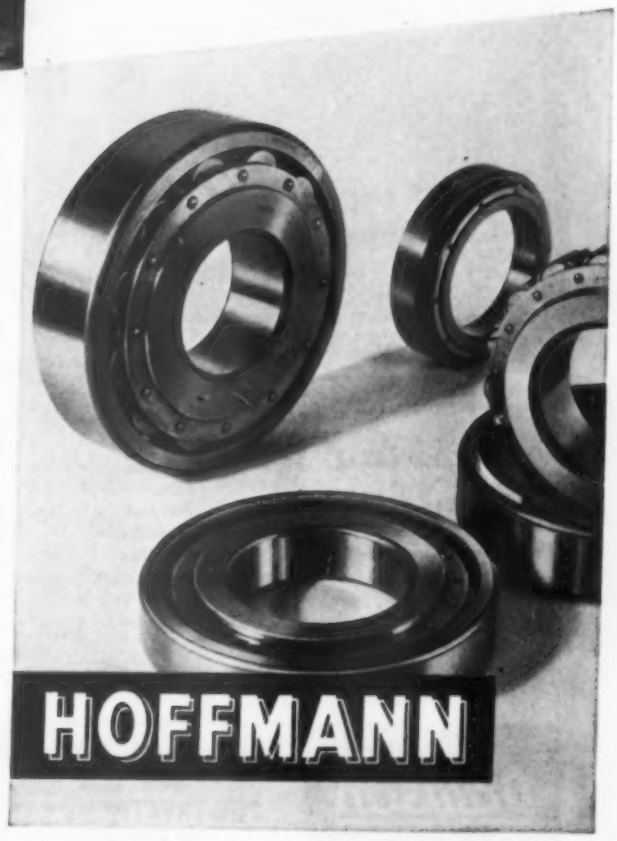

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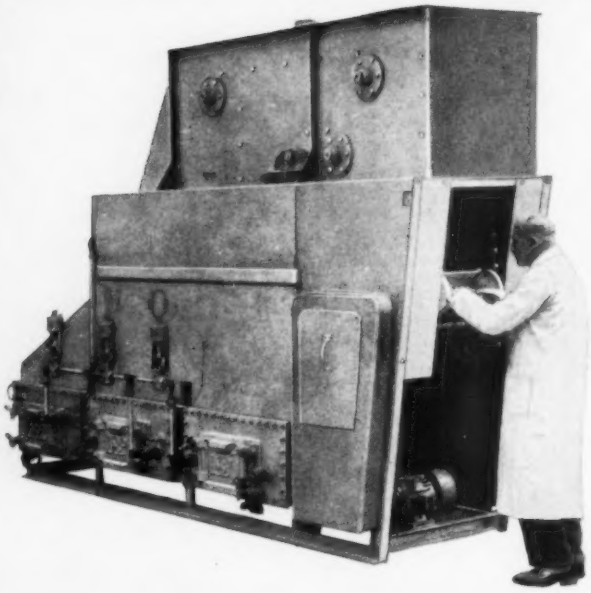

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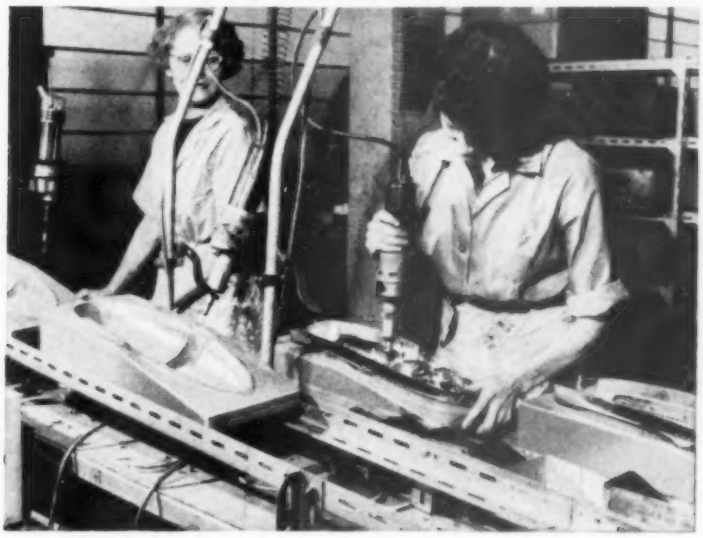

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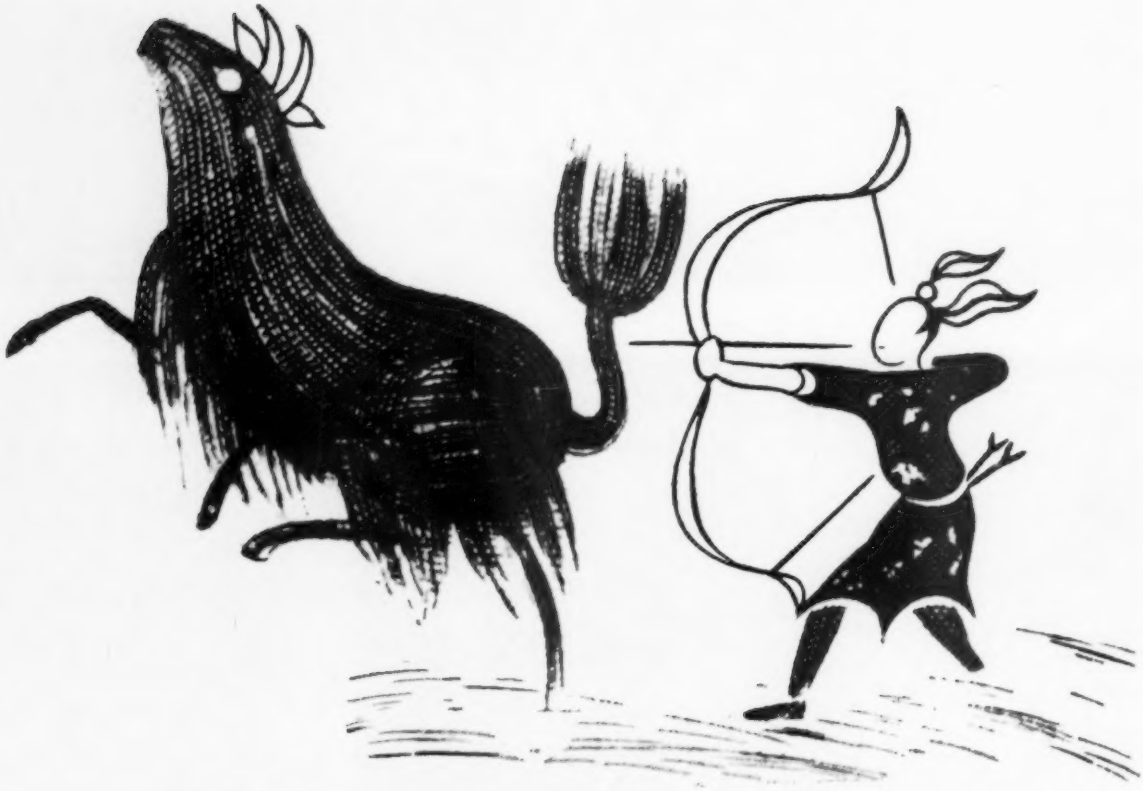

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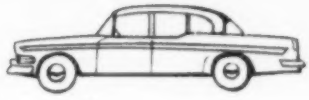

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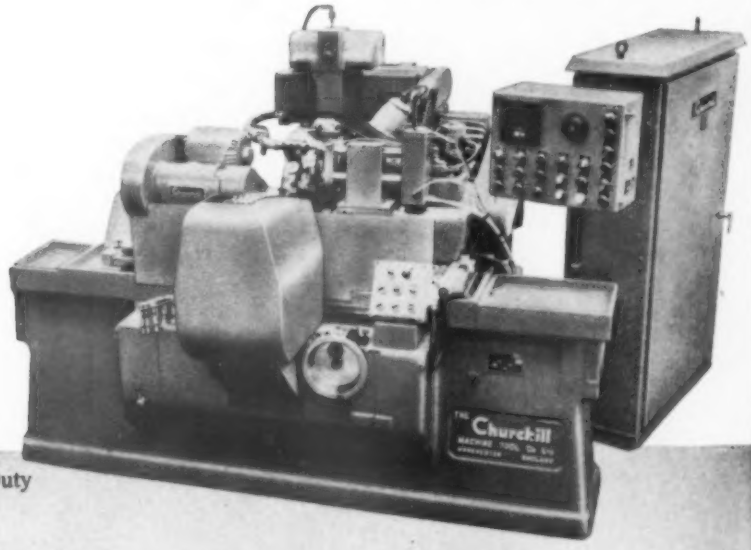

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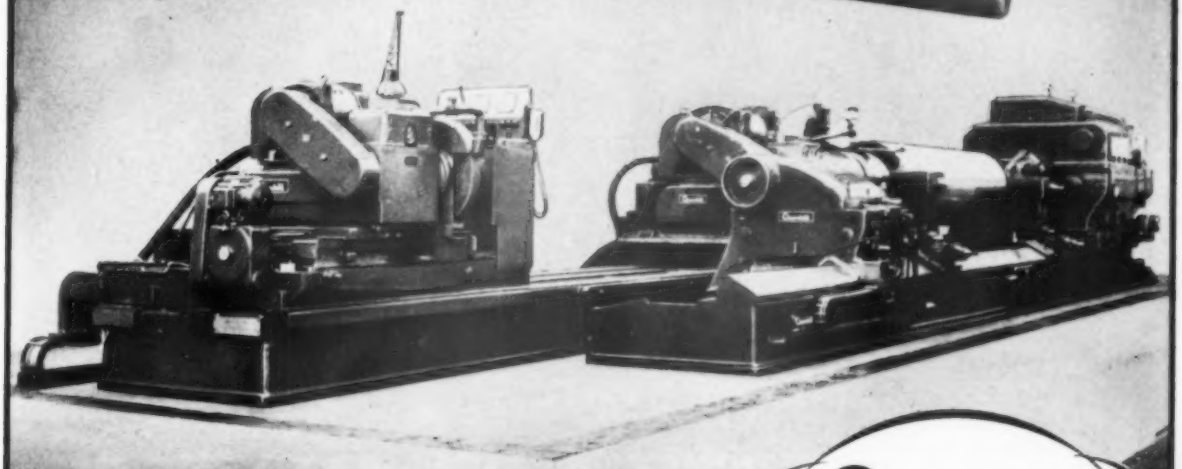

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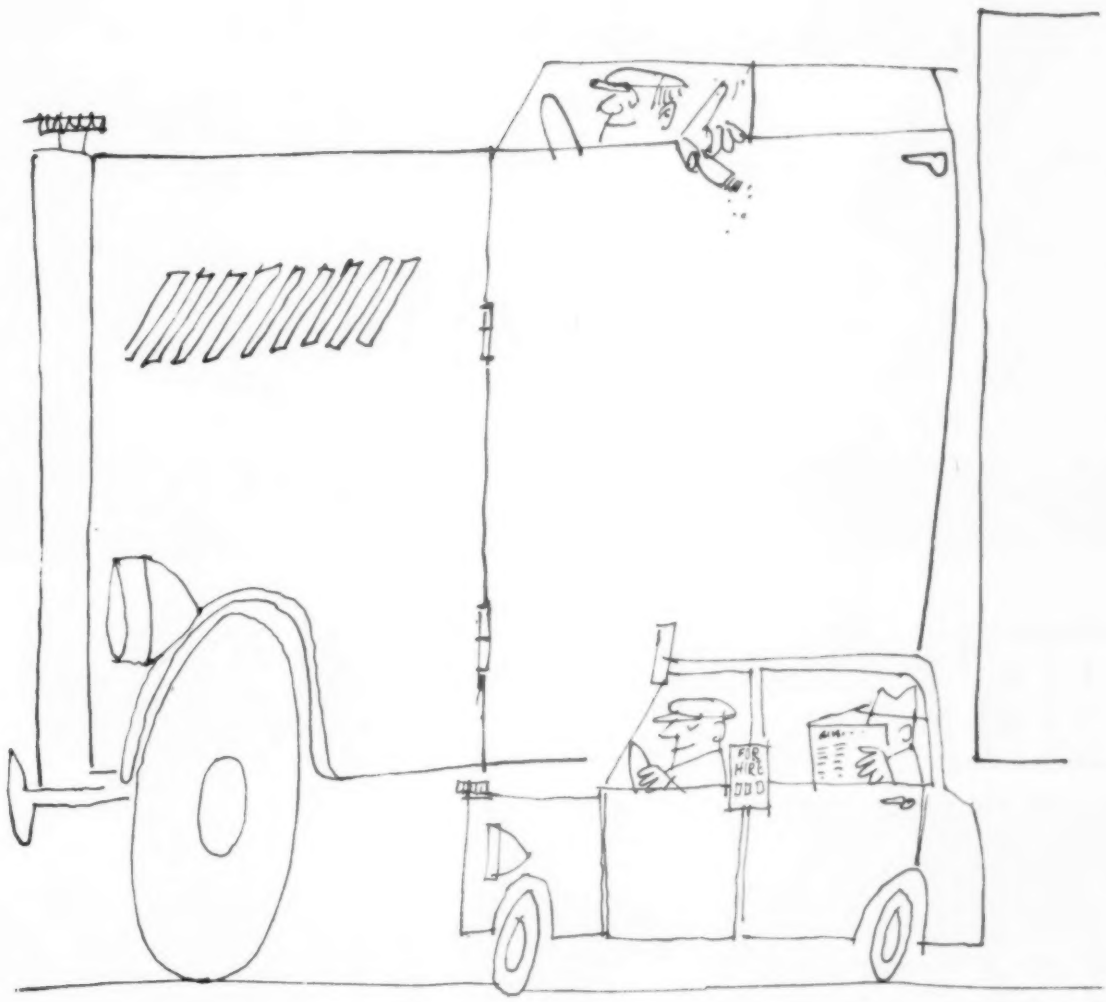

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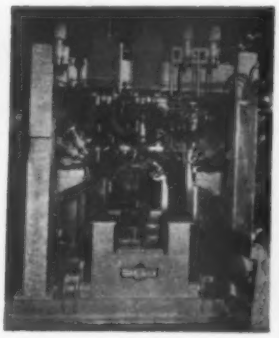

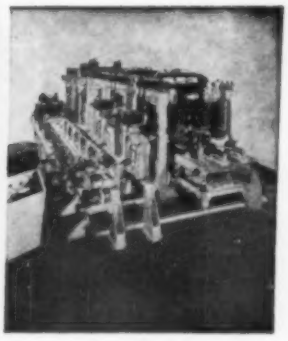

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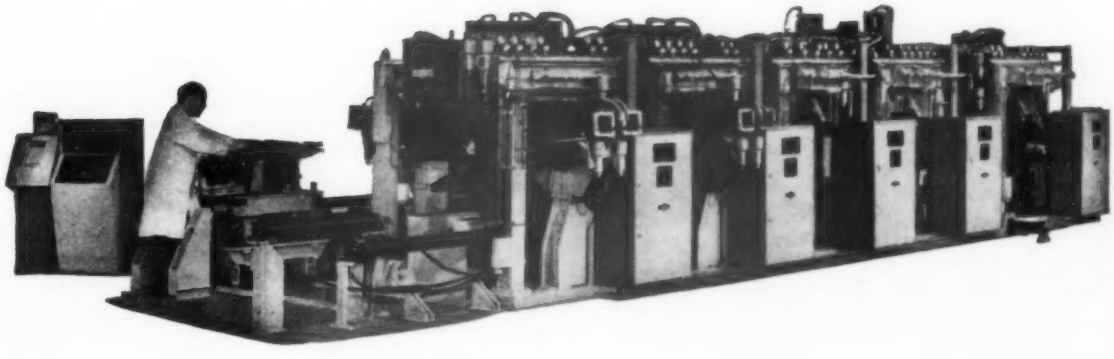

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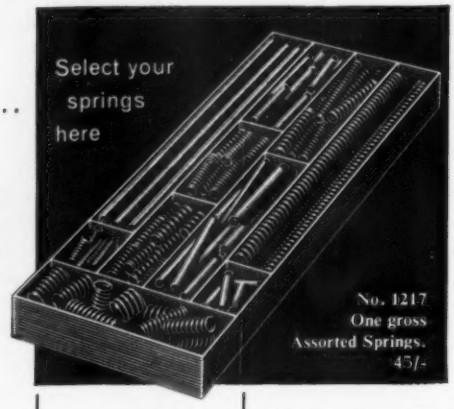

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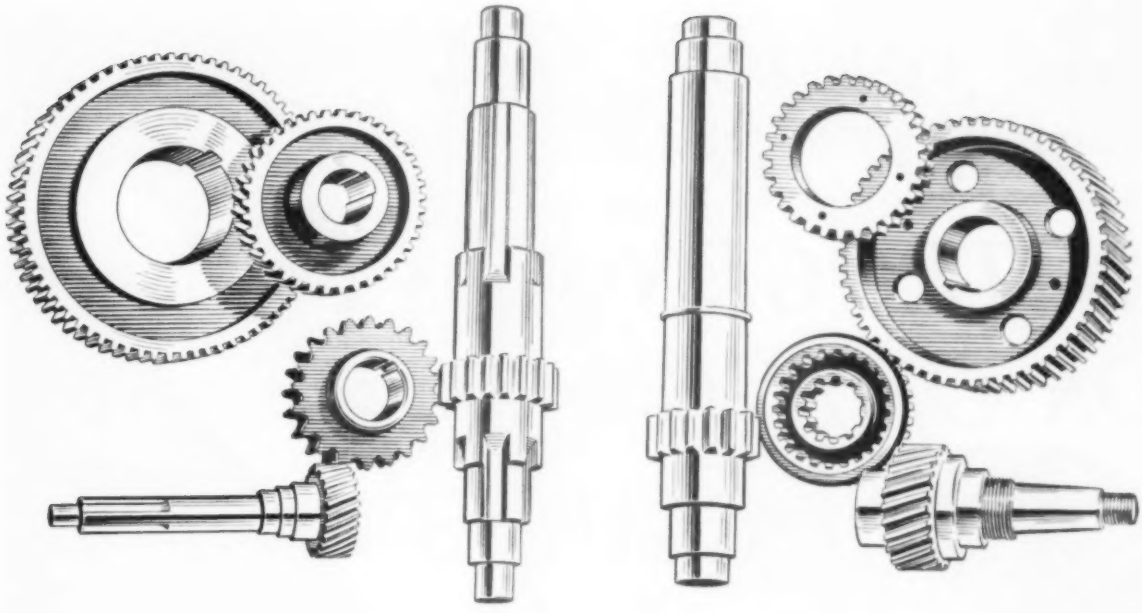

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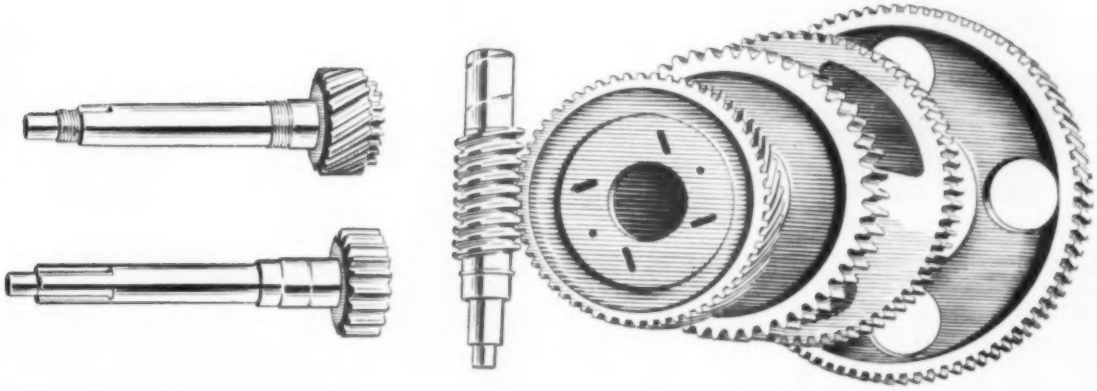

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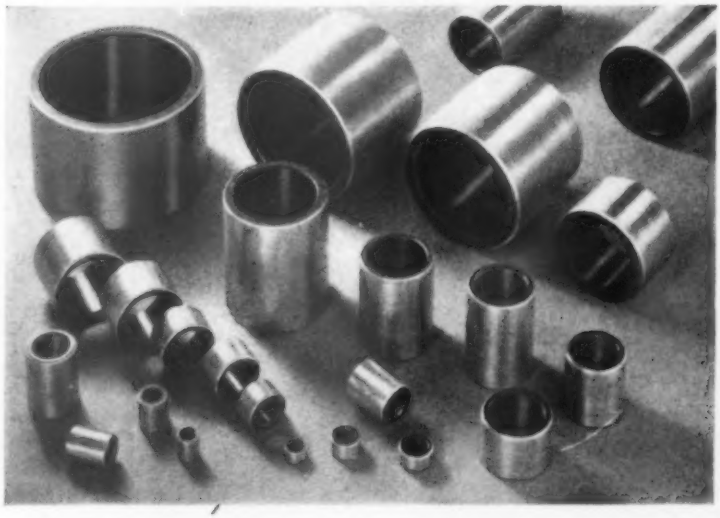

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